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The Trend of Air Transportation

By EDMUND T. ALLEN,[1] SEATTLE, WASH.

The year 1925 marked the beginning of air transportation as an industry. Since then it has advanced through successive stages of growth and development until today as the author believes air transport is in the state of transition between the pioneering period and that of mature growth. Airway mileage by scheduled air transports in the United States has increased from a total of 2,000,000 miles flown in 1926 to 90,000,000 in 1939. Air passenger-miles in 1938 aggregated 600,000,000. In this paper the author reviews the technical developments in aircraft and improvements in airway operation which have made possible this phenomenal growth. Every phase of this development which has played a part in the successful and safe operation of the airway systems of the present day is treated comprehensively in order that an understanding may be gained of the future possibilities of air transport and the lines along which it will advance.

AIR TRANSPORTATION has arrived at its present state of commercial success in the space of a very few years. So sudden has been its development that the public has hardly been able to keep pace with it, or to understand it, or to accept it fully, although certainly patronage of this mode of travel is steadily becoming more and more general.

To the outsider, who catches only the high lights and does not see the effort and meticulous research that make possible the spectacular developments, the advance of aviation and of air transportation has been one series of sensations after another. No sooner has the news of one innovation cooled than another startling announcement supersedes it. In other words, it is an industry which, because there has been so much to be accomplished, has been moving ahead not at a walk but at a forced run, just as fast as engineers could make it move and as fast as air traffic could pay for it.

Air transportation as an industry started scarcely fourteen years ago, with a long, long way to go. Scarcely stopping to consider what the ultimate goal might be, those responsible for its being proceeded on their course with the idea of finding out more about that goal along the way. Literally, they looked to the sky as the limit.

How far has air transportation progressed along its course? What are its present status and its future prospects? Most new industries, that are sound, first go through a period of growth and then arrive at a state of maturity. The first stage is one of rapid change and development, merging into the second, which is one of refinement and the perfection of detail. In which stage does air transportation find itself today? Or is it at the merging point between the two periods?

The engineer likes to make graphs and study curves. When they begin to flatten out, he feels that some sort of limit or goal is being approached, be it temporary or permanent. He knows that the effort must be greater for further accomplishment and gain. Is the curve of efficiency in airplane performance flattening out? How about the curve of airplane reliability?

A study of the records of the last fifteen years and of the present trends of the industry gives some enlightenment on those questions. The author believes such a study will show that, parallel to the development of other forms of transportation, air transport has reached the transition between the pioneering period and the period of mature growth. It will also show that the optimum airplane is closer at hand; that, having reached a certain leveling out in performance and size of aircraft, the industry is concentrating now more definitely upon the finer problems of perfection in safety, comfort, and reliability.

Review of Recent Air-Transport Developments

One of the best ways to judge the future trends of any industry or to see the present direction of its development is to look back a short distance into the past of that industry. In such a review, it is necessary to go back far enough to get beyond the seasonal or short-swing tendencies. Fortunately, in the air-transportation industry, it is possible, in the space of a few years (1925 to 1940), to view the entire history of a major development, its early faltering steps, its seasonal ups and downs, even its decline in a major economic depression and its partial recovery toward a normal level.

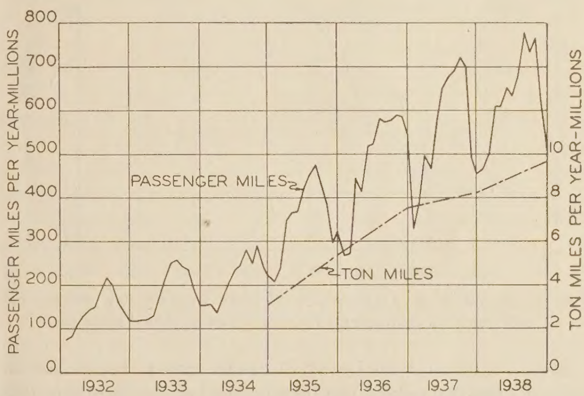

Fig. 1 Passenger-Miles and Ton-Miles Flown on United States Domestic Air Lines

To look back over the record of the past through the eyes of the traffic man is most inspiring, especially when it is considered that the figures are merely an indication of what may be expected of the future. Air passenger-miles in the United States have risen from 120,000,000 in 1932, fairly steadily except for regular seasonal winter declines, to a total of 600,000,000 in 1938. There was a slight dip in the rising trend when in 1934 the number of passengers carried actually decreased as compared with the number in the preceding year, but this was due primarily to the depression. Cargo transportation by air, principally in the form of air mail, has similarly increased in ton-miles flown. The rate of this increase has been approximately 50 per cent per year, Fig. 1.

In some other transportation fields, it is true, the growth has been equally rapid. Rail transportation grew more rapidly than air transportation during its early boom, and bus transportation

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has also had its period of tremendous expansion. But neither of these had to overcome such a serious reluctance on the part of the public to accept a mode of travel so new and different—one that took them out of their accustomed element and into the air. Both of these other modes of transportation had heavy indirect subsidies, such as land grants and highway construction. Aircraft transportation has had similar assistance of federal and municipal funds in airway and airport construction. Beacon-lighted routes have been built, forming a network over the entire country for night flying, and radio-range routes now permit the radio navigation of any aircraft with a radio receiver.

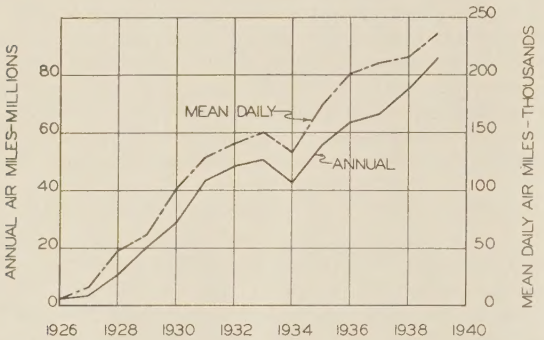

Fig. 2 Annual Airway Mileage and Mileage Flown Daily on United States Domestic Air Lines

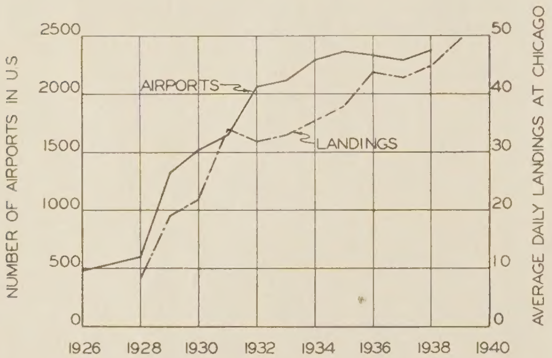

Fig. 3 Number of Airports in United States and Scheduled Air-Line Landings per Day at Typical Air Terminal

Airway mileage in the United States has grown from an annual total of 2,000,000 miles flown by scheduled air transports in 1926 to 10,000,000 in 1928; 30,000,000 in 1930; 50,000,000 in 1933; and 90,000,000 in 1939. The year 1934 shows a characteristic depression dip in the curve. The curve for mean daily air-miles flown closely parallels that for the gross annual mileage, as shown in Fig. 2.

The increase in the number of airports has been an important trend, perhaps not closely related to long-distance, scheduled, air transport, but an indication of the growing interest in and use of aircraft by private owners, which occurred principally between 1926 and 1932. The 500 airports of 1926 had increased to 2100 in 1932. It would be expected that eventually there would occur a leveling out in the increase of the number of airports. This has definitely occurred during the last four years.

The number of landings per day of scheduled air-line transports at a typical air terminal may also be used as an index of the trend. A curve of landings, Fig. 3, based on studies made at the Chicago air terminal, rises sharply from ten in 1928 to 34 in 1931 and then drops with the depression, rising again in 1936 to a peak of 44. The subsequent drop is definite evidence of the increase in size of the aircraft unit, which factor in itself would cause a decrease in the number of such units landing per day at an air terminal.

Technical Advance of Transport Airplanes

As in air traffic, so in the transport airplane itself, the progress of fourteen years has been tremendous. During the period from 1925 to 1939, the transport plane has changed from a small single-engined biplane, carrying 2 passengers in addition to the pilot, weighing 1 ton gross, and costing less than $10,000, to a large monoplane with either 2 or 4 engines, carrying 30 to 70

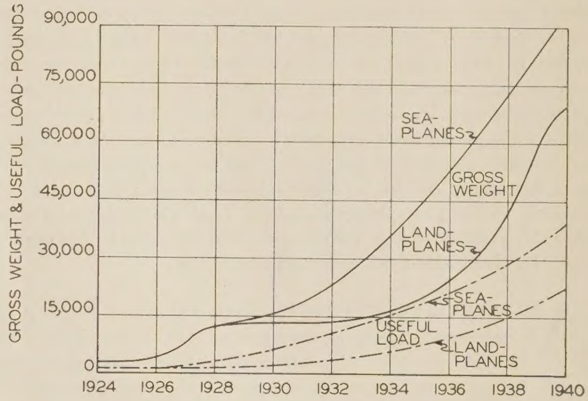

Fig. 4 Gross Weight and Useful Load, United States Transport Airplanes

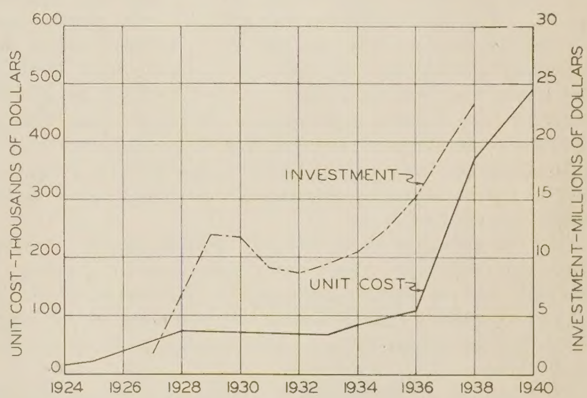

Fig. 5 Unit Cost of Typical Air Transport and Investment in Transport Planes, United States Domestic Air Lines

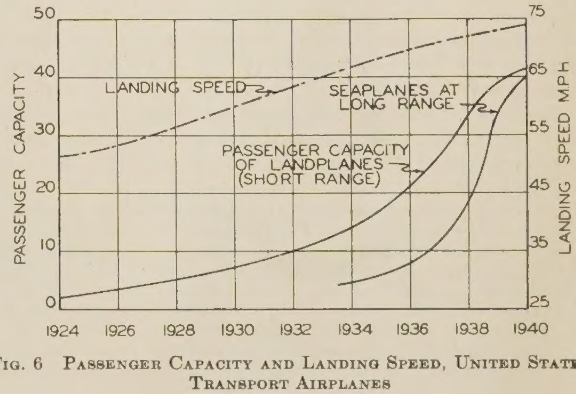

Fig. 6 Passenger Capacity and Landing Speed, United States Transport Airplanes

passengers plus a crew of 4 to 11, weighing not 1 ton but from 20 to 40 tons gross, and costing $250,000 to more than $500,000 each, Figs. 4, 5, and 6. The increase in size has been very gradual and has been dependent at almost every step upon parallel technical development. Each forward step has been based on the successful performance of the preceding one. Each small advance required a successful experimental and design venture in order to obtain the necessary continued influx of working capital to finance the expansion.

Domestic air lines had a total investment of less than $2,000,000 in transport airplanes in 1927. This grew to $12,000,000 in two years, sank for five years during the depression, but recovered quickly and rose to $23,000,000 in 1938. The unit cost of a typical transport airplane in 1927 was $23,000. The curve of increasing unit cost, Fig. 5, shows a gradual increase up to $110,000 in 1936 and then a large and sudden increase as the 4-engine air transports were developed. Cost per pound of useful load shows the effect of the demand for luxury of travel. In 1928–1929, $12 per lb represented the cost; in 1939 it is $22 per lb.

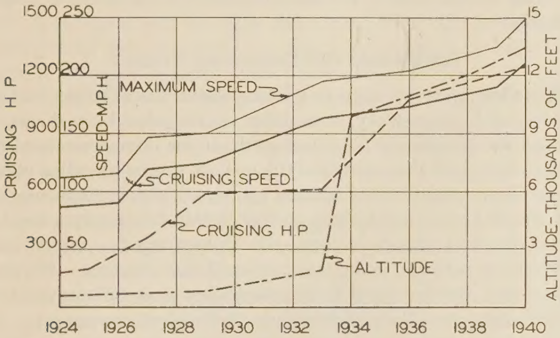

Fig. 7 Cruising and Maximum Speeds; Typical Altitude and Cruising Horsepower per Airplane, United States Domestic Air Lines

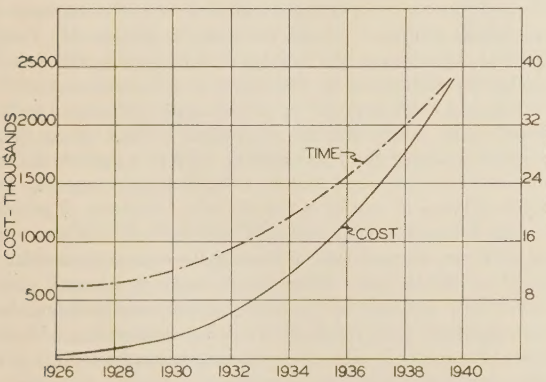

Fig. 8 Cost and Time of Developing a New Air-Line Transport

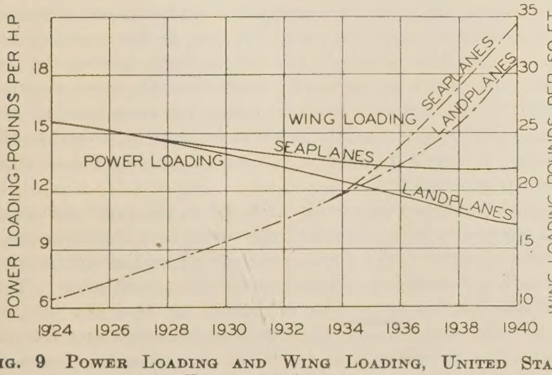

Fig. 9 Power Loading and Wing Loading, United States Transport Airplanes

The average transport airplane of 1924 had a cruising speed of 100 mph, using from 200 to 300 hp and flying at a cruising altitude of less than 1000 ft. The 1940 airplane, equipped with a pressurized cabin, cruises at 200 mph, using cruising horsepowers of 1200 to 2000 at altitudes ranging from 12,000 to 20,000 ft, Fig. 7. These changes have not been uniform or gradual, but have occurred in sudden spurts, as new engines, new techniques, and new design tendencies were developed.

One of the most interesting trends in this connection is that of the enormously increasing cost of developing a new air-liner design. In 1927 this averaged $40,000, while today costs as great as $1,500,000 are not uncommon. It now requires not six months as it did in 1926, but two years to produce the first airplane of a new major type, Fig. 8.

The trend of air-transport design toward larger size and higher power output has a very interesting parallel in the trend toward increasing wing loadings and decreasing power loadings. The average 1924 transport airplane had a power loading of 16 lb per hp and a wing loading of 11 lb per sq ft. Wing loading, going up from a low value, and power loading, going down from a high value, crossed in 1935 at approximately 13 lb each. By 1939 the average power loading of landplanes had gone down to 10 and the wing loading of landplanes up to 30. The combined loading has thus through the years shown a gradual increase from 27 in 1924 to 35 in 1935, and to 40 in 1939. Wing loading is going up faster than power loading is coming down. This indicates the increasing efficiency of modern design. Seaplane wing loadings are definitely on the increase and will probably pass the 50 mark soon.

It would be expected that landing speeds would show a very large increase parallel to the increased wing loadings, and this is true to a certain extent. Higher lift devices, however, have made it possible for designers to increase gradually the weight carried per square foot of wing area without greatly increasing the landing speed, Fig. 9. Wing loadings of long-range aircraft no longer are an indication of landing speed, because now such aircraft are not landed with the take-off wing loading. Provisional gross weight on take-off is reducible by dumping extra fuel in case a landing is required soon after take-off, before this fuel has been consumed in flight. Thus 70 mph is required for landing at standard gross weight, but take-offs may be made with greatly increased loads. Landing speeds of 55 mph were typical in 1924 to 1927, while landing speeds of 70 mph have become fairly well standardized at the present. There is evidence however that they will soon be on the rise again with the advent of stable landing gears and smooth runways.

Government regulations have for years limited wing loadings of seaplanes as well as landplanes to the point where 70-mph alighting was possible. The perfectly apparent safety, however, of much higher speeds on touching the water or land with aircraft of 40 tons gross weight, and the great increase in economy and range available by increasing the wing loadings, indicate that such an extreme limitation is no longer essential.

Development of Safety Features

Improvement of the airplane, its equipment, and its operation from the standpoint of safety has been the most wholesome and most reassuring trend of all. Much has been accomplished already, and much more is now being done.

The addition of proved safety devices has been a very gradual process. Outstanding among these have been blind-flying in-

struments and equipment. The gyroscopic turn indicator was the first standard equipment to be developed, but technique in its use was not required by government regulations until twelve years after the instrument was adopted. Improved aperiodic compasses, directional gyroscopes, and radio communication were necessary before extensive blind flying could be safely carried out as a scheduled operation. The automatic pilot, which still further aids blind flight, is not permitted on tests for pilot perfection in the technique of blind flying, orientation, and navigation, all of which must be done with only the compass, bank-and-turn indicator, and radio range.

Blind-landing systems, long considered by pilot and management alike as a hazard rather than a means of promoting safety, are now regarded almost uniformly as one of the last remaining steps to be taken in order to achieve safety from all that class of hazards deriving from unpredictable weather. Severe wing icing, excessive propeller icing, and extreme static of radio disturbances still remain primary problems which are now receiving the greatest attention of engineers and research staffs.

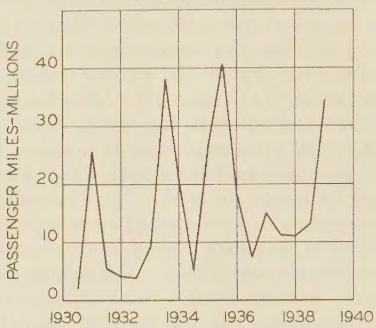

Fig. 10 Passenger-Miles Flown per Fatality on United States Domestic Air Lines; Semiannual Totals

The governmental requirement for multiengined passenger airplanes placed a restriction upon the use of single-engine aircraft and focused the attention of designers upon the problems of flight characteristics after an engine failure. Early multiengined airplanes were incapable of continued flight with full load after an engine failure. On later designs, it was specified that there must be no point in the flight range at which the airplane will become dangerous should any engine fail. This called for new criteria for control and stability, new research in wind-tunnel and aerodynamic theory. It has resulted in greatly increasing aircraft safety by virtually eliminating accidents caused by a failure of the power plant. It has of course increased the cost of aircraft and somewhat decreased the economy of operation because of the requirement for a greater excess of power available per passenger carried.

In operating practices, the trend through the years of air-transportation development is both interesting and significant, particularly with respect to a noticeable change in pilot attitude and management policies. There has been a maturing soberness in the gradual elimination of piloting exhibitionism, and the development of sound safety policies in the conduct of transport air lines. The curbing of the desire to stunt airplanes was not very evident until 1931, when both management and piloting personnel became acutely aware of the economic necessity of eliminating the spectacular from the lures offered to the public to travel by air. The air lines which first made air passengers realize that there are no sensations to air travel, that there is never any excitement and that the pilots are not daredevils, were the first to build up passenger business. The multimotored airplanes of 1928 started off the acceleration toward conservatism, but as late as 1931, loads of passengers were occasionally stunted in airplanes. In some South American countries, this evidence of the infancy of an industry still exists.

This trend points to a maturing of both management and pilots. The older management policy of chance taking in combatting bad weather was typical of a passing stage of development. "The mail must go" slogan cost the lives of 4 out of 48 air-line pilots each year before passenger flying dictated a sounder policy. Without blind-flying instruments and without radio, the air-transportation industry was born and struggled through its early developmental stages. During those stages, the courage of the pioneers did much to build toward a scientifically sounder future but that future has no place for the early type of pioneer airmen. The air line which encouraged chance taking in 1929 has, if it survived at all, become the acme of safety in operation in 1939, Fig. 10.

Lightening the Personnel Burden

To follow up this change in piloting attitude and policy, in the interest of further safety, some little scrutiny has been directed toward the question of the relationship of the human mechanism at the controls of the aircraft to the tasks it was being called upon to perform. Analyses of human reactions, under conditions of stress such as fear and fatigue, indicated that frequently a breakdown occurred simply because the tasks were too many and difficult to perform under such a combined loading. Physiological and psychological tests indicated a vast differential between pilots as to their performance under these conditions.

New techniques of careful control of the engine and equipment operation, the great increase in the number of controls and instruments, the introduction of radio and its resultant radio-navigation problems, all of these new additions to the already overloaded human mechanisms required a new envisagement of the personnel problem. Could the tasks be lightened? Could the general physiological and training level be very greatly raised? Could testing techniques be developed to determine accurately the rating of a pilot required to perform such varied and multitudinous tasks under difficult conditions? Work along these lines resulted during the years 1930 to 1939 in a gradual annual improvement in personnel and in a slight improvement in the stability and ease of control of the aircraft. Selection of personnel changed from the older notion of choosing only the man with many years' experience to one of training the man along standards designed by the air line. Once the personnel is selected, training continued not only in the sensorimotor coordinations, but also in theoretical study of aerodynamics and engineering. Health was cared for methodically and psychological analyses were periodically made to determine whether any factors of worry or of mental stress were being introduced to weaken the pilot's preparation for his task.

Copilots at the same time changed in status to one of almost equal importance with the pilot. In fact, as the tendency has been to train first pilots via the copilot route, leaving the old-timers who could not compete mentally with their younger brothers in the captain's position, a situation arose in which the young copilot was frequently found to be more capable than the captain. It was a question of specialized training versus trial-and-error experience.

Accident analyses frequently have led to the conclusion that, in an emergency when several things went wrong simultaneously, the plan of action broke down, especially where the captain tried to handle all decisions and frequently all controls alone. This possibility has led to the plan of drilling the crew as a unit for emergencies, and through such drilling to foresee any possible combination of emergencies for which a solution has not been worked out. When this is done thoroughly, emergencies cease

to exist; they are relegated to the routine for which there is a ready solution.

Development of Cruising Control

The most striking change in technique of air-line operation, from the standpoint of its economics and likewise from its effect on safety, has been the development of the theory and practice of cruising control. Prior to 1934 most aircraft operation was done at low altitude, and cruising-power output was approximated for sea-level conditions. The result was that very little was known by the pilot as to the optimum cruising operation of his engines. Most flying was done at absurdly low power with occasional high output operation at sea level where damage was done which caused engine failure or increased maintenance costs. The automatic power-and-mixture control on one air line and, on another, the development of cruising guidance charts rather suddenly changed all this, increased the scheduled cruising speeds of air transports some 20 mph, raised the cruising altitude, and, at the same time, virtually eliminated the dangerously damaging operation which had been making engine maintenance costs so high. The advent of the controllable propeller made this operation possible by permitting a change of pitch from the take-off condition, where high power output was necessary at low altitudes, to the cruising range, where low power output was necessary at moderate revolutions at high altitude. Optimum cruising from the engine-airplane viewpoint was at high altitude at exactly desired cruising brake mean effective pressure. The utilization of the conception of optimum cruising altitude and the whole technique of optimum flight path followed closely upon this development. Cruising control at exact mixture conditions then became a reality.

These technological developments occasioned the rather sudden increases in cruising speeds of 1934 to 1936 and ushered in the conception of substratosphere cruising in pressurized cabins where passengers and crew alike can breathe comfortably and not become fatigued at high altitudes.

Today exact control of all engine operation is now regarded as requisite on all air lines. Contact flying near the ground is now restricted to the end of the optimum flight path. Meteorological and radio improvements have assisted in making this most desirable technique possible.

Present Requirements in Transport Aircraft

The record of the past—of the increase in air-traffic volume, the radical improvement of the transport airplane, the progress in safety and in operating practices—is quite clear. What then are the present requirements of the transport plane, and what are the trends of present development? The question covers a great deal of territory—size and range, speed and general performance, comfort and safety provisions, reliability. An analysis of each of these is necessary to clarify the important question as to what the air-line operator—and the public—requires and expects of the modern transport plane, and how and to what extent these expectations are being realized.

The special problems of the air-transport industry arise largely from the economical demands for services. These demands have frequently been dictated or guided by government regulations. They are largely affected by the popular appeal and are quite sensitive to the injurious effects upon the industry of aircraft fatalities. In general the primary demand is for through-service with relatively few stops. There is at present practically no demand for limited and local services, largely because the competing rail and highway services very nearly equal in speed and comfort whatever the air lines can offer, after getting passengers out to an airport and in from an airport at the two terminals of a short-distance trip.

The designer's problem in meeting the demand for these through-express services with large pay loads is to provide the range plus the pay load plus the passenger and express capacity required for the route to be covered. Since 1932 virtually all transport airplanes have had a fuel capacity far in excess of that which could be filled if a full pay load were also carried. To a greater and greater extent, since then, airplanes have been designed to carry either of two possible useful loads; a large pay load for a short range with most of the fuel tanks empty, or a relatively small pay load for a long range.

The practice on the air lines in loading at terminals has been to refuel only after the pay load was known and then only up to an amount which gave the maximum allowable gross weight, which ordinarily meant that the fuel tanks were from one half to two thirds full. This amount of fuel must of course equal the minimum required by considerations of safety for the trip contemplated. The type of aircraft which could be used economically on either long trips or over long stretches of impossible landing weather, or else on short trips with a very heavy pay load, gave great flexibility to the operation.

The demands of operators continue to require such flexibility in ever-increasing degrees. As an example it is desirable to have an aircraft capable of carrying 30 to 40 passengers from New York to Washington in 1 hr and then to be able to take off with 20 passengers for Miami, with ample fuel for a 2000-mile flight in case of emergency.

For overseas routes, the demand is now crystallizing around a large flying boat of from 30 to 100 passengers capable of nonstop flights of 3000 miles. Longer range than this proves to be unnecessarily costly because of the excessive fuel loads required and the resulting increase in power required and in cost of operation. The trend has definitely established the 20- to 40-ton-aircraft size for overseas service, whether of the seaplane or the landplane type. Because of the virtual elimination of forced landings in multiengined aircraft, landplanes of multiengined varieties are being considered very seriously for overseas routes because of the greater economy of flight due to drag reduction.

For overland services, there is appearing a stronger demand for a smaller landplane in addition to the type now being developed in the 30- to 50-passenger class. Cost of operation remains almost constant for any given scheduled service whether the passenger seats are filled or not. Thus it becomes economically unwise for the route on which density of traffic is light to equip with aircraft costing \$1 to \$2 per mile to operate, when such a route can be equipped with small aircraft costing 30 to 60 cents per mile. The cost of operation remains the same for full as for empty passenger planes.

There is no consistent demand for cargo carriers. From time to time, air freighters have been developed and cargo services inaugurated but these have not lasted long. The laws of the land have been built upon the indirect subsidy and encouragement of combined passenger and mail and cargo transport to such an extent that a service for cargo only cannot survive against competition offering cargo plus passengers. This fact has led designers away from cargo-only types and toward a flexible unit capable of transporting economically a wide variety of loads but definitely having seats for enough passengers to pay for the service in case other forms of cargo fail to materialize. Most domestic air lines depend upon their air-mail contracts in addition to passenger-and-express revenue in order to break even. The effect upon design criteria of requiring such a variety of cargo capacity for a given fixed useful load indicates the probable increasing utility of large landplanes for both overseas and overland routes.

The determination of the possible economical use of a new aircraft is a very complicated procedure. Other things being equal, the percentage of useful load to gross weight would be a determining factor in estimating economy of operation. This percentage

of useful load to gross weight increased rapidly from the early stages of aircraft construction in 1910, until approximately the time of the birth of air transportation as an industry in 1925. At that time it had reached approximately 50 per cent; that is, half the gross weight of the airplane was structure and power plant and the remaining half useful load. Since then, the demands for increasing comfort, increasing safety, increasing speed, and increasing general utility, have far outweighed the improvements in structural efficiency and also the improvements in aerodynamic efficiency. Luxury features have become so important that the percentage of useful load has now dropped to approximately 33 per cent from the 50 per cent figure of 1925.

The Trend in Speed

Requirements for increasing speed in transport aircraft have been decreasing during the last few years. Cruising speed at a given cruising altitude is, of course, not the speed which can be scheduled for any operation. Point-to-point speeds or, as they are called in this country, "block-to-block" speeds, are the speeds which interest the operator. These are very much lower than cruising speeds because of the time required to maneuver for a take-off, climb to cruising altitude, descend from cruising altitude (a positive gain), maneuver for landing, and taxi after landing. Block-to-block speeds are less than cruising speeds by an amount depending primarily upon trip length; for very long flights they approach equality, but for short flights they may vary as much as 20 per cent. Thus a 200-mile flight between airports made in an aircraft capable of cruising at 12,000 feet altitude at 200 mph would require not 1 hr, but approximately $1^1/_2$ hr (50 per cent more) because of the loss of time required for climbing up to the cruising altitude and the time used in take-off and landing. On a 600-mile flight the scheduled speed can come within approximately 10 per cent of the cruising speed.

Although any further increase in speed is beneficial, speeds have already reached the point where they require less emphasis; and more emphasis is currently being given to the matters of comfort, safety, and economy of operation. However, in forecasting the possible future demands for aircraft speeds, it is of interest to note that the primary consideration has now become the possibility of "saving a business day." This is particularly important on routes having heavy traffic density where convenience, in the sense of frequent schedules, and availability to businessmen mean money saved.

Ordinarily it would be thought that transcontinental routes across the United States would be the only ones where such a saving would be possible. An examination of the air schedules in many parts of the United States reveals, however, that this possibility of saving a business day can be realized on some relatively short trips as well as long ones. There are many north-to-south routes and many diagonal trips, involving connections between two air lines, which become of great importance in this matter of saving time.

Now that air-line transportation has reached the 200-mph cruising stage, a very large increase in speed would be required in order to expect a saving of a business day on any given trip. Strangely enough, the coast-to-coast route, westbound, against the prevailing wind, is now being made overnight; whereas the eastbound trip with the prevailing wind requires the better part of a business day in addition to the overnight time. This paradox is explained by a 6-hr daylight differential between the east-west and the west-east trips in crossing three "time" zones. A typical west-east schedule leaves the west coast at 5:00 p.m. and arrives on the east coast at 12:00 m. To save a business day would in this case require arrival at 9:00 a.m., with 3 hr less flying time. To accomplish this would require an increase in block-to-block speed from 173 to 216 mph, or approximately a 25 per cent increase, assuming that the same number of landings would be made en route.

The most obvious way of increasing speed of air carriers is to increase their cruising power. This is also the most expensive way, since an increase of approximately 3 per cent in power is required for a 1 per cent increase in speed.

This increase in power involves tremendous increases in fuel capacity, fuel weight, and, of course, fuel cost. It involves also large increases in the weight of the power plant and in structural weights required to carry these additional weights.

A much more economical means for increasing cruising speed is an increase in cruising altitude, whereby drag is reduced because of the decreased density of the air. Ten years ago, however, transports had already reached altitudes as high as passenger comfort would allow. It has only been within the last year that designs have become available permitting higher cruising altitudes without passenger discomfort. The trend in this direction, for long routes, seems to point toward still higher cruising altitudes up to 20,000 ft or even a possible 25,000 ft for transcontinental service.

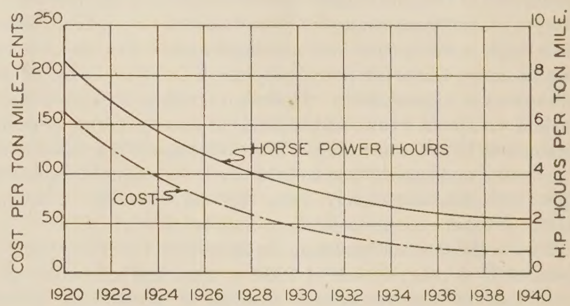

Fig. 11 Horsepower-Hours per Ton-Mile of Pay Load and Cost per Ton-Mile at Cruising Speed, United States Transport Airplanes

Speed increase by means of parasitic-drag reduction has been the stand-by of all the airplane improvers for many years. This possibility has been much overworked especially for the layman. Drag reduction is very difficult indeed to achieve and is very costly in structural design. It is safe to say that there has been no easy road as yet found, despite recent press releases, for appreciable reductions in aircraft drag.

It is of great interest to note that the curve of horsepower-hours per ton-mile of pay load at cruising speed has been gradually flattening out for United States transport aircraft during the last four years, and it seems very unlikely that there will be any appreciable decrease in the amount of work required to transport a given cargo by air at a given speed for the next ten years, Fig. 11.

Much of the subject of airplane speed and performance and, for that matter, airplane reliability as well, is concerned with the power plant. What are the status and the trend in this important phase of the transport airplane?

Developments in Aircraft Engines

American aircraft engines have only recently begun to utilize piston speeds near 3000 fpm for maximum-power operation. British practice has considerably exceeded American values in the past, although their engines have not been operated at brake mean effective pressures as high as is the practice here. The net effect of these differences has not been large when measured in terms of power output per unit of size. However, the higher pressures used in this country have required a very sturdy engine structure, which has been quite readily adapted to the increased rotative and piston speeds now being utilized. This change has

been accomplished with only moderate increases in the weights of engine components, and a small net reduction in weight per horsepower has fortunately been achieved. Some engines are now available which develop 1 hp for about 1.1 lb, including propeller-drive reduction gearing. It seems unlikely that this figure will be much improved for current basic types of engines without a prohibitive sacrifice in reliability.

The widespread use of gasoline having a high-knock rating has been a primary factor in recent steps toward improvement of the aircraft engine. The first improvements, achieved by careful choice of the base crude oil, selective fractionation, and addition of tetraethyl lead as a knock suppressor, provided a foundation for major advances in the engine-power ratings. The gains thus initially achieved have stimulated the study of properties of a wide variety of synthetic fuels. This work has now resulted in the use of blends of natural gasoline having strictly controlled properties and synthetic hydrocarbons having a high order of resistance to detonation. This practice has removed some technical barriers to still higher engine performance and there is ample indication that the limit of such improvements is still in the future. The mutual adaptation of engines and fuels has been materially advanced in this country by cooperative research in which all interested agencies actively participate.

Use of fuels of high-octane value has not only permitted large increases in maximum-power rating for take-off, but it has had an important and favorable effect upon engine durability and upon cruising-power output, largely by eliminating detonation hazards at normal cruising powers with the leanest practicable fuel-air mixtures.

With the current trend to higher flying speeds in transport operation, there has been much concern for the drag of the large radial engines. Conventional values for cooling drag can be reduced drastically by air-flow controls which proportion the pressure head and the amount of air flowing over the cylinders to the existing need for cooling at any particular operating condition. The power used for cooling of cylinders may be held within limits heretofore thought practicable only for liquid-cooled engines. For extremely high speeds, however, the liquid-cooled engine appears to have a definite advantage over its competitors, largely because of more favorable shape and proportions. Speeds used in air-transport operations scarcely can be expected to rise in the near future to values where the low-drag shapes, associated with liquid cooling, will bear sufficient premium to offset the operating advantages of direct air cooling.

There appears to be a trend in aircraft engines toward the development of two principal variations of basic engine types. One of these variations provides a maximum of power output for high-performance aircraft at some sacrifice of reliability and fuel consumption. The other variation is constructed to provide a minimum fuel consumption for long-range operation with a slightly reduced maximum-power capacity. This trend is highly significant to air transportation because of effects on the economics of long-range operations.

Since the utility of an air transport (particularly for long-range service) is primarily dependent upon its pay-load capacity, every other weight involved in the airplane structure can be evaluated in terms of dollars per pound. The gross weight chargeable to the whole propulsion system consists of the engine, the propeller, structural supports, cowling, accessories, plumbing, etc., plus fuel and oil required for the schedule. A study of pertinent data shows that fuel and oil weights are a predominant influence in this figure for nonstop flights of more than a few hundred miles. A typical air-transport engine will consume a quantity of fuel and oil about equivalent to its own weight in 4 or 5 hr of operation at the maximum power required for cruising. Thus for a 20-hr flight, weight savings on the fuel consumed are about four times as important in terms of pay load as are items of fixed weight, such as the basic engine. For the same reason, any items affecting the propeller efficiency at cruising power are extremely important.

In the selection of engines for long-range transports, engine factors which are conducive to a maximum of safety and reliability are, of course, of primary importance. Ease of maintenance, accessibility in flight, and proper adaptation to service requirements are much more important factors than in the case of other aircraft types. A premium is placed upon experience accumulated in previous operations of a similar type. It is essential that the aircraft shall have serviceability characteristics which will eliminate need for extensive attention between scheduled flights. As aircraft become larger, this is a factor of increasing importance because of the larger investment and consequent overhead charges.

These factors emphasize the necessity for a degree of reliability in the long-range-aircraft power plant which approaches that provided in marine practice. It seems probable that the future will show an increasing emphasis on durability and reduction in fuel and oil consumption at moderate power-output levels. It is fortunate that all concerned have come to realize that these qualities cannot be obtained without moderate sacrifices in the high performance qualities associated with other classes of aircraft service.

Provisions for Comfort of Passengers

The progress of air transportation depends in very large measure upon passenger comfort and passenger satisfaction. Volume of traffic is very sensitive to improvements in these two factors. The air lines in the United States have been very responsive to these demands on the part of the traveling public and, in turn, they have demanded of the aircraft designer more space per passenger, less crowding, better seat and berth design, larger dressing rooms, better ventilation and temperature control, air-conditioning, and, now, air-pressure control for high altitudes and to minimize the effect of rapid changes in altitude. In addition to these demands, improved food service aloft is now required, such that appetizing meals can be served attractively and expeditiously. Improved lighting for passengers' accommodation in reading and writing has become such a serious problem that in some cases auxiliary power plants have been required to supply the current demands for luxury items as well as mechanical necessities. The weight of these luxury items has increased on the average from 10 lb per passenger in 1930 to approximately 180 lb per passenger in 1939. Floor area has increased from approximately 5 sq ft per passenger in 1932 to 20 sq ft per passenger in 1939, and cabin volume from 25 cu ft in 1931 to 120 cu ft in 1939. This increase in roominess is in itself a primary comfort feature, Fig. 12.

Soundproofing has changed from a luxury item, only a few years ago, to a primary requirement today in crew and passenger quarters alike.

The increased effort to cater to passenger satisfaction has led

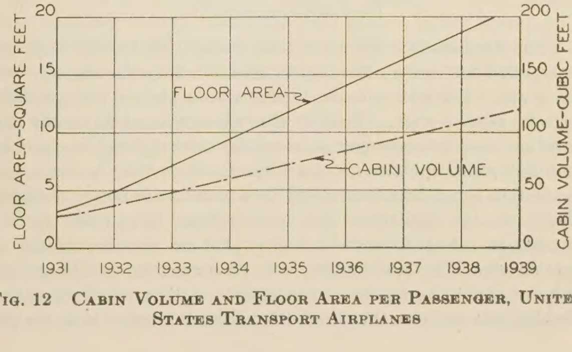

Fig. 12 Cabin Volume and Floor Area per Passenger, United States Transport Airplanes

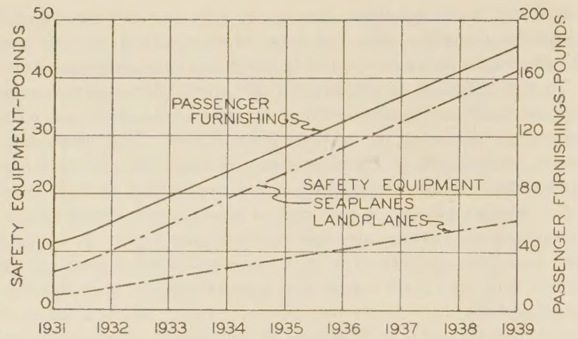

Fig. 13 Weight of Safety Equipment and Furnishings per Passenger, United States Transport Airplanes

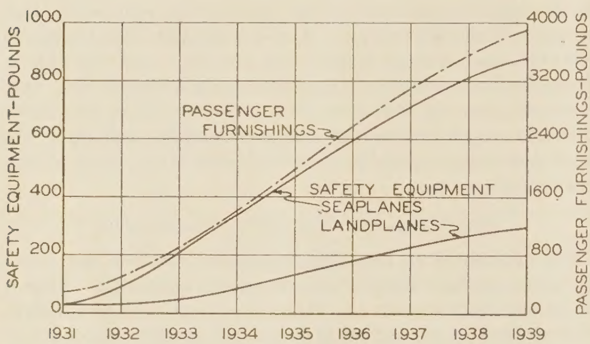

Fig. 14 Weight of Safety Equipment and Passenger Furnishings, United States Transport Airplanes

designers to develop many new devices and facilities. The desire to make the passenger wholly comfortable has also directed attention to the smoothness of the flight path as a vitally important factor in the pilot's technique, as well as in the designers' repertoire. Modern aircraft flies very much more smoothly than that of ten years ago, partially because of the greater altitudes for cruising, partially because of the increased stability of the airplanes, and partially because the pilots are endeavoring every moment to eliminate any flight irregularity which would cause passenger discomfort. The fact that a large number of transport schedules are now arranged for sleeping passengers has accelerated this development of smoothness of the flight and has tended to reduce the number of stops for fuel.

Great care is now taken to change altitude slowly because some of the passengers may be distressed by rapid altitude changes. The trend in this direction points definitely for nonstop services for most of the sleepers. This tendency will require larger fuel loads, greater weights of luxury equipment, and definitely higher costs per passenger-mile.

Demand for Safety

There has definitely been a radical change in the attitude of air-transport lines toward the importance of safety during the last few years. It is now considered an intimate part of the specifications for new aircraft. The fact that a single accident results in a great decrease in passenger revenue, far outweighing any gain in economy which could be obtained by omitting safety features, has resulted in the demand for safety on a scale never before realized. Governmental regulations and requirements have built up this interest in safety to such a degree that no possible source of trouble should be overlooked, and no structural or equipment features should be considered negligible, if they can decrease in the slightest measure the possibility of an accident. How far the industry has gone in this direction can be visualized by a glance at the increasing weights of such safety equipment during the last few years. In 1931 the average seaplane in the United States had approximately 50 lb of safety equipment, whereas today the average seaplane has 850 lb of such equipment. In weight per passenger of safety equipment, the increase has been equally rapid for our seaplanes, rising from 10 lb per passenger in 1932 to 40 lb per passenger in 1939. On landplanes the increase has been 300 per cent in the last seven years, Figs. 13 and 14.

In listing the requirements for an air-transport airplane in a highly competitive field, various factors have assumed, from time to time, the leading place. Safety, as such a feature, is sometimes assumed to be in the "of course" category. In other words, it has been assumed that the airplane must be safe. Grammarians may quibble over whether safety is a relative term; aircraft operators know it definitely as a feature which must be weighed carefully against other features. Conceivably, an airplane might be loaded with safety equipment until it could not get off the ground, or at least until it could not carry any pay load. Those who make a realistic and rational approach to the problem will recognize that there are certain hazards to flying operations, just as there are in any other form of transportation, and that we can only hope to decrease these hazards gradually during the advance in technological development.

Any such advance is made at the cost of economy; the four-engine airplane for instance is definitely less economical than the two-engine airplane but, if it is capable of continued flight after any two engines have failed, it is some 27 times more reliable than the two-engine airplane, based on actual probabilities of one engine failing. It is not safe, however, to consider a hazard as being reduced 50 per cent merely because the system involved is duplicated. There are certain types of failure of an engine, for instance, which will cause an accident regardless of the number of engines, unless the potentialities of safety involved in this duplication are fully realized by making each unit duplicated merely an incidental factor for continued flight.

There is a continual argument among engineers regarding such duplication of vulnerable parts versus improvements to reduce vulnerability. Both trends will probably continue side by side in aircraft development.

One of the most persistent problems of aircraft design is the element of human error of the operator of the aircraft. As with vehicles of land and sea, this problem can be solved by simultaneous attacks in two directions: (*a*) Improvement in the quality and condition of the operators to reduce to a minimum the mistakes they will make; and (*b*) design of controls and the aircraft's response and stability in such a manner that mistakes, delays, or other errors will have a very minor effect upon the operation. Examples of this kind are to be seen in all modern aircraft design. Landing gears are now designed so that they absorb energy without bouncing the aircraft into the air again when an error in judging distance from the ground results in contacting the ground with a considerable vertical velocity or with an angular velocity such that the wing's angle of attack is increasing. Ground-looping was a serious problem requiring skillful handling before the design improvements recently made resulted in elimination of this hazard.

The most serious errors in flight technique frequently result from forgetting or omitting some essential adjustment prior to take-off. The complexity of large aircraft can be visualized after examination of the pilot's "check-off" list where from 10 to 20 items are specifically to be checked, (1) prior to take-off, (2) prior to landing, or (3) before any change in operating regime. The failure to check off this list has resulted in accidents simply because it is impossible for the human mechanism of the pilot to retain at the focus of attention the number of items necessary.

Sooner or later, in repeated operations, especially when the pilot has been under strain for a long period, some essential item will be forgotten. Designers are now bending all their efforts toward (1) making such omissions of minor importance to the safety of the aircraft and (2) providing automatic devices to make the omissions impossible.

Although large aircraft utilize a crew of from three to seven men, great effort has been expended in retaining complete control of the aircraft under one man. Such a one-man control is necessary only in emergencies where the time required to coordinate the efforts of the crew members may not be available.

Automatic Controls Promote Safety

The trend toward automatic devices continues to point to the errorproof control. One direction in which this type of design has developed is a light signaling system which permits one man to check all items requiring his attention by a momentary glance at a bank of red and green lights. Prior to take-off he presses the take-off button, whereupon any condition of the airplane, unsuitable for take-off, at once causes a red light to flash on his warning board. Each regime of operation has its corresponding push-button, which automatically flashes the red or green stop-and-go lights, proper to that regime. Such automatic warnings of pilot errors have been in use for many years, but they have frequently failed to prevent the error, sometimes calling attention to omissions after it is too late. The more recent developments in this field are promising.

In this country, there is an almost universal use of automatic pilots to relieve the pilot of the strain of attending to the actual mechanism of flight. Pilot fatigue has been studied under many conditions of flight, particularly those involving mental and emotional strain. Any relief from fatigue improves the pilot's judgment.

On aircraft where the power-plant control is in the hands of the pilot, a large part of his attention is required to maintain optimum conditions of power and mixture and revolutions for the flight regime in which he is operating. Automatic-mixture controls for carburetors are now considered a "must" on air lines. Automatic power control is rapidly coming under this category. Ice-free carburetion has virtually eliminated the need for control of carburetor-air temperature. Automatic control of propeller pitch or engine speed is now standard on all air lines. There is some indication that detonation indicators will soon be perfected in response to the demand for a positive indication of this damaging phenomenon on engines in which the cooling is so excellent that it is no longer a guide to detonation. Some operators feel that the trend toward automatic devices has gone too far, and the statement is frequently heard that an extra man is required in the crew to see that all the automatic devices work. Those of us, however, who have followed these same developments in rail transportation, will note a great similarity both in the trend toward automatic devices and the resistance to them.

The problems of air-line operators as related to the characteristics of aircraft, during take-off and landing, have occasionally conflicted with the ideals of the government regulating bodies. In general, however, government regulation has definitely acted as a steadying influence to prevent the sporadic introduction of seemingly attractive aircraft having questionable safety. Since the government furnished the airports, in most cases, and also the airway aids to flying, it was of course quite logical to expect governmental regulation in establishing minimum take-off and landing performance.

Take-off run in itself, or what is sometimes called unsticking distance, has been established for many years as 1000 ft. Obstacle clearing is now recognized as a far more important criterion of safety in take-off than the unsticking distance itself. Either of these criteria, when established as minimum requirements, virtually determines the power-weight ratio of the aircraft design. It is very interesting to note in this connection that, although all transport aircraft must demonstrate their ability to unstick within the required 1000 ft, such a take-off is never used in air-line operations. The technique of operation on air lines requires that the airplane be held on the ground until it reaches a certain air-speed exceeding by a definite margin the minimum speed for take-off. In some cases this requirement is considered as a sort of safety factor to prevent inadvertent stalling after take-off. In the case of multiengine aircraft, it has another specific usefulness, namely, to make certain that, in case any engine fails during take-off, the airplane will be under good control. Until recently, the speed for minimum control with one engine inoperative, has been considerably in excess of the speed for minimum control under symmetrical conditions of power. The gap between stalling speed and minimum speed for control under the worst unsymmetrical power condition, was accepted as the hazard which must accompany the advantages of multiengine operation. However, recent improvements in aircraft design have resulted in a narrowing of this gap to the point where no aircraft in operation will ever run the risk of having an engine failure on take-off throw it out of control.

The landing-speed requirement virtually establishes the wing-loading of modern aircraft used for passenger transportation. Maximum-lift coefficients of acceptable wing-flap arrangements are virtually all the same, for modern wing designs. A standard of 70 mph is now established in this country as the landing speed of aircraft above 20,000 lb gross weight. With wing-loading and power-loading thus determined by governmental regulation and also by the acceptance standards of the major air lines, transport aircraft are tending toward greater and greater identity of general design.

Flight characteristics, after a failure of part of the power plant during take-off, are of vital concern for the safety of air-line operation. The take-off has generally been considered a less dangerous or critical part of the flight, than the landing. It is now being recognized as a much more critical part. Even though an airplane can be demonstrated to have control with the worst unsymmetrical power condition, at a speed close to a stall for this power condition, it has been found in practice that frequently the pilot has been unable to handle the emergency of an engine failure during take-off. Possibly the delay in the application of corrective measures resulted in allowing the airplane to begin a turn, and in this turn the airplane was uncontrollable by reason of a yaw in the "wrong" direction. At any rate, the critical conditions of flight in this emergency made any failure of the pilot at any one of a number of important points too serious to overcome. The margins for safety at this point must thus be larger than at any other point during flight.

Major improvements in stability are still being sought by aircraft builders. Longitudinal stability is at present required only to the extent of a damping of dynamic oscillations. Increased static stability is very desirable, both from a safety standpoint and also for comfort and freedom from pilot fatigue. Improvements in yaw-and-roll stability are proceeding in the direction of attaining the proper coordination between the slopes of the yawing- and rolling-moment curves. The designer's problem of obtaining satisfactory yaw stability at small angles of yaw is still a major one. Spiral stability, although not ordinarily required, is a desirable flight characteristic. On large aircraft, it assumes an entirely different aspect than it ever had on small aircraft, where the pilot could only detect its presence or absence by a carefully conducted test, with controls free. On aircraft above 30,000 lb, controlled flight comes to resemble free flight more and more. On such an aircraft, in which spiral stability was determined to be

negative, it would be clearly evident to the pilot in making a turn, such as an approach to a landing, that the forces required to prevent the airplane from tightening up in the spiral were higher than they should be for both comfort and safety.

Aircraft Maintenance Problems

The air-line problems of maintenance have demanded and obtained such highly competent technicians during the last few years, that the entire subject of air-line engineering has achieved a new importance. One of the most interesting comments upon this development is the change in the relationship between piloting personnel and maintenance personnel. At first the maintenance personnel was the key to the problem of reducing mechanical failures. Very soon, as this reduction in failures proceeded, it became evident that certain pilots were very hard on equipment and certain other pilots, who apparently had maintained just as good schedules, never had any mechanical failures. Analysis of the operating technique disclosed what practices were causing most of the failures, and what types of operating procedure resulted in trouble-free running.

Maintenance personnel is interested primarily in insuring safety from mechanical failure, rather than in the repairs or deterioration of equipment. The parallelism may be noted here between preventive versus corrective measures in this field and in medicine. The most successful maintenance man is the one who never has to maintain anything. Aircraft inspection practices have now reached a very highly perfected state on most air lines. Airplane major and minor inspections and overhaul are now done primarily by checking an outlined form. Engine inspection and overhaul proceed along a similar routine path, each valve, each piston ring, each aileron, and each inner tube has its individual record of hours of operation and date for next replacement.

The maintenance of safety devices has become a very important feature of the overhaul shop; a safety device is of course not a safety device if it is not maintained properly. All equipment such as radio, for example, receives its separate maintenance procedure and replacement quite regardless of the fact that it may be functioning perfectly at the time of removal. Instruments are now checked in air-conditioned and well-equipped laboratories. Flaps and airplane controls all receive separate and special attention, each with its individual trip check and routine time checks after a certain period found by experience to be best for each individual item. Deicer boots have been a very serious problem for maintenance engineers, until some means was devised to prevent their deterioration and static punctures. They still remain a rather costly item for maintenance in winter.

There has been in the past a great deal of competition between air-line maintenance engineers; secrets found by one group were carefully guarded to prevent another group from equaling their record of maintenance perfection. Some years ago, however, the air lines realizing the enormous importance of bringing all services up to the standard of the best, joined their forces and pooled their information in a series of semiannual conferences, at which they discussed freely all problems relating to air-line-engineering practices. These conferences resulted immediately in a marked reduction in mechanical failures throughout the entire United States. All air lines benefited from them, even the best ones receiving a stimulation to greater effort by the contact with the poorer services.

The Outlook

The trends of the present period show two distinct and quite opposite tendencies in the development of air transportation. One group of factors, such as size of airplanes, speed (at sea level) and economy of operation, shows a tendency to level out, as if a kind of limit were being approached. In the case of flying speeds it must be noted, however, that further increases are being obtained by virtue of operation at higher altitudes with pressureized cabins. Aircraft size has reached its leveling-off stage because of economic considerations. Technological development is at the stage where airplanes twice as large as any flying today could be built, but such airplanes could not meet the economic problem of making a profit in air-line use with any type of service demanded today. In regard to the other leveling-out tendencies, such as sea-level speed and economy, a great deal of effort bent on improvement along these lines yields relatively little further gain. It may well be that some new development will suddenly reverse the trend and shoot us all into high gear again along these lines, but the chances are that the next ten years will show no more increase in these factors of speed and economy than the last five years. In this category come also factors such as landing speed and work done at cruising speed per ton-mile of pay load, thus indicating a leveling out in airplane efficiency.

The second group of factors shows the opposite tendency. These curves are not tending to level out but rather to increase in slope. The industry's efforts, responsive to this divergence in tendencies have recently been directed from the less promising to the more promising trends. We are now accelerating the development of safety and reliability, of comfort and convenience of travel, with cost of airplanes on the upswing. Altitudes of flight are pointed upward, with pressurized cabins almost certain to dominate the field within the next few years. This means in itself greater comfort for the passengers, greater safety above the weather, greater range to eliminate intermediate refueling stops, greater reliability of schedules, and greater flying speeds.

Where the graphs do not in some cases bear out the conclusions, the reasons may lie in the fact that the points plotted were so scattered that, especially at the 1940 end of the curves, the trends were left up to the judgment of the plotter who could have varied the lines up or down a great deal with equal justification from the available data. The curves show averages, but averages do not always show trends. Another factor which may need elucidation at this point is the confusion which results from trying to average the 1938 airplanes of the twin-engined 20,000-lb variety, and the 1940 airplanes just now entering the field with four engines and 40,000 to 60,000 lb gross weight. These latter represent 1940 more truly than the smaller airplanes, even though in 1940 there will still be many more of the smaller ones in use than the larger ones. The curves in many cases average between these two, whereas the true story might have been told more clearly with broken lines leveling off at the 1940 stage.

The first trend toward leveling out occurs in those factors always present in the initial stages of an industry and incident to its greatest period of growth. That we are approaching limits here, even though temporary ones, indicates the coming of the age of air transportation. The second trend, because it occurs in such factors as safety and comfort items is also confirmation of the maturity, the end of spectacular pioneering, and the application of sound economics to the problems of broadening the foundation of the industry.

Discussion

E. F. Burton.[2] The writer has had ten months of fast-moving aviation development as an advantage in discussing Mr. Allen's paper. Viewed at the time of preparation Mr. Allen gave considerable advance thinking; viewed today he is found partly right and partly wrong due both to domestic and international events of importance definitely influencing otherwise normal development.

[2] Chief designer and assistant chief engineer, Douglas Aircraft Co., Inc., Santa Monica, Calif.

It is difficult to agree with Mr. Allen on two major contentions: First, that his curves and conclusions drawn therefrom on gross weight plotted against time are indicative that 60,000-lb aircraft will supplant the lighter types, and second that "there is no consistent demand for cargo carriers." Majority opinion is now confident that the larger-capacity aircraft will merely complement existing equipment for the more heavily used routes and that only a very large increase in passenger traffic will render obsolete the aircraft in the twenty-five to thirty-five thousand pound gross-weight class. This is not anticipated for many years.

Mr. Allen has proved by his own statistics that safety and passenger comfort provisions have required the loss of many valuable pounds of pay load as well as increased costs. It follows that these two vital fundamentals raise the cost of transporting goods to a point where such carriage require lower initial and operating investments and special handling of aircraft freight. The term "aircraft freight" is used necessarily since aircraft can never compete with railroad mass transportation.

Aside from these two arguments Mr. Allen should be congratulated for his clear analysis of existing conditions, the developments leading up to the present time, and the glimpse given us on things to come.

J. P. Van Zandt.[3] Referring to the curve of Fig. 11 which shows the cost per ton-mile at cruising speed for United States transport airplanes, will the author explain his method of computing the values? The average cost per capacity pay load ton-mile of the domestic air-mail carriers was about 50 cents for the fiscal year, ending June 30, 1935, whereas, the curve of Fig. 11 indicates that the lower limit of 20 to 25 cents had been approached by that year.

Another question which arises is whether or not Fig. 12 refers to seaplanes only? The curves shown seem excessive for Douglas planes or for the Boeing 307.

Also referring to Fig. 12, does the author really mean that "the cabin-volume curve represents landplane practice?" His figure for cabin volume per passenger for 1939, on a steadily rising trend, is about 120 cu ft. The writer's understanding, however, is that the cabin volume per passenger in the Boeing 307 is between 50 and 55 cu ft and approximately the same in the DC-3. This fact appears to be borne out in a recent paper by Bonnalie of the United Air Lines.[4] The writer has always assumed that 50 to 60 cu ft per passenger was ample cabin volume for landplanes rather than 125 cu ft, as the curve in question evidently predicts.

The figure of 28 cents per pay load ton-mile capacity for the DC-3, based on a 1200-mile range, is of interest, but further information on the method of computation used should be given. The operating cost of any type of plane depends to a large extent upon how it is used. The number of miles flown per plane per year, for example, directly affects all the costs related in any way to the utilization factor. If in one instance the overhead and administrative costs are $50,000 a year and in another instance $250,000 a year, for the same number of total ton-miles flown, then the overhead costs per ton-mile in the latter instance are 5 times that of the former. But this relation would no longer hold if the number of miles flown per plane per year were different in the two cases.

Author's Closure

Referring to the question concerning Fig. 11, of the paper; the author believes that the discrepancy between his figures of cost per capacity pay load ton-mile of the domestic air-mail carriers and the figures cited by Mr. Van Zandt may be due to the "range" for which the capacity pay load is computed. The author's calculations were based on a minimum range and maximum pay load condition. It would be logical to find these figures doubled, if based upon a maximum range condition. A figure of 28 cents per pay-load ton-mile has been attained for the DC-3, based upon a 1200-mile range. The author would appreciate further data on this matter if it is available from sources at command of the discussers.

Fig. 12 represents something of a compromise between seaplanes and landplanes. Admittedly the justification for presenting the case in this manner is slight, because of the appreciable differences between the two classes of aircraft. Unquestionably, separate curves would have been more desirable. The cabin-volume curve represents landplane practice rather than seaplane practice. The floor-area curve at the 1939 point approaches seaplane usage. If landplane practice concerning floor area had been consistently followed, the curve would have tapered off to a value 20 per cent lower than the one shown.

[3] Technical Consultant, Civil Aeronautics Authority, Washington, D. C.

[4] "Toward Economic Air-Line Equipment," by A. F. Bonnalie, Trans. A.S.M.E., vol. 62, Jan., 1940, Fig. 2, p. 3.

An Improved Technique for Centrifugal-Pump-Efficiency Measurements

By ROBERT W. ANGUS,[1] TORONTO, CANADA

A duplicate water-supply system has been recently constructed by the City of Toronto, Canada and, in connection therewith, an extensive pumping station was built at Victoria Park. This station contains eleven centrifugal pumps, driven by induction and synchronous motors, the capacities of the pumps ranging from 6,000,000 gal per day to 60,000,000 gal per day, and the heads ranging from 54 ft to 270 ft. All of the pumps are guaranteed for very high over-all efficiencies, some of them to over 86 per cent, and the penalties for failure to meet the guarantees are very high. The tests on them, therefore, had to be made with unusual accuracy. The paper describes the methods adopted and the photographic recording of the observations, together with results of some of the tests.

THE City of Toronto, situated on the north shore of Lake Ontario, has always drawn its water supply from the lake. Two intakes have long been in use for delivering water into wells on Toronto Island, from which it is pumped to the filtration plants. The filtered water then passes through a tunnel under Toronto Bay to the main pumping station in the city at the foot of John Street; from this point it is distributed to the various sections of the city. To facilitate this distribution, auxiliary pumping stations have been installed at strategic locations, and reservoirs have also been built.

A few years ago it was decided to supplement the supply by constructing a new intake some miles east of the original location, and to provide in connection therewith an independent pumping station and filtration plant, together with other necessary additions, including a new reservoir on St. Clair Avenue. The new pumping station is located at Victoria Park on the Lake Shore, and contains the pumping machinery with which this paper deals.

At the present time, this station contains eleven pumps, but space has been left for the addition of two more at a later date. Of the pumps now installed, three are intended to deliver wash water to the filters, four others are for the purpose of lifting the raw water as received from the intake onto the filtration plant, and the remaining four pumps deliver the filtered and treated water to the distribution system. However, the St. Clair Avenue reservoir floats on the main delivery line so that surplus water flows into it. The three wash-water pumps have capacities of 6,000,000 gal, 9,000,000 gal, and 12,000,000 gal, per 24 hr, respectively, at a net pressure of 54 ft, while the four raw-water pumps are rated, respectively, at 24,000,000 gal, 30,000,000 gal, 48,000,000 gal, and 60,000,000 gal per 24 hr at a net head of 82 ft. Of the four filtered-water pumps, two are for 6,000,000 gal each per 24 hr at 196 ft net head, while each of the other two has a capacity of 30,000,000 gal per 24 hr at 270 ft net head.

Each pump is direct-connected to a General Electric motor, but the switching equipment was supplied by the Canadian Westinghouse Company. All motors operate on 25-cycle 3-phase 2300-v current. Induction motors of 75 hp, 100 hp, and 140 hp, with a rated speed of 730 rpm have been installed on the three wash-water pumps. All other pumps are driven by synchronous motors, the 48,000,000-gal and the 60,000,000-gal raw-water pumps operating at 500 rpm, while the remaining raw-water pump and all four of the filtered-water pumps operate at 750 rpm. The motors on the raw-water pumps have rated outputs of 425 hp, 525 hp, 825 hp, and 1025 hp, respectively, while the filtered-water pumps are driven by 250-hp and 1700-hp motors. Synchronous motors are supplied by exciters running at 125 v direct current and are designed for 90 per cent power factor.

Acceptance Tests on Pump Installation

The principal data for the pumps are given in Table 1. The sizes of the suction and discharge pipes were taken as the average across the two diameters which terminated in the piezometer openings. The efficiencies are the over-all results as between horsepower of the electric input to the motor and the horsepower in the water delivered by the pump, the figures tabulated being those guaranteed by the builder in each case. All other values in Table 1 are from the specifications.

The pumps and the piping connected with them were built and installed by the Dominion Engineering Works, Limited, Montreal, Canada. All are double-suction pumps without guide rings; the filtered-water pumps are in reality two single-stage pumps, in each case coupled in series by piping. Therefore, all of the pumps are of the same type and class. The specific speeds are calculated from the formula

$$N_s = \frac{N\sqrt{G}}{H^{3/4}}, \quad \text{and} \quad N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

where N is the speed in revolutions per minute, H is the head per stage in feet, G the gallons per minute rated output, and Q is the equivalent cubic feet per second.

The specifications for the pumps were carefully drawn up, giving full details regarding the acceptance-test procedure, describing the methods of measurement, and the points of attachment of the gages. In addition, it was stipulated that sums ranging downward from $4450 according to the pump were to be deducted for each 1 per cent that the over-all efficiency failed to attain that guaranteed; these large deductions made it necessary that the greatest accuracy should be maintained throughout every stage of the test.

When the author was instructed by R. C. Harris, Commissioner of Works, Toronto, to carry out the acceptance tests on the machinery, the various parts of the installation were in process of erection and readily accessible. All of the suction- and discharge-nozzle diameters were measured with micrometer calipers so that the proper areas for calculating velocity-head corrections might be known; also the upstream and throat diameters of the venturi meters were measured on two diameters, the upstream sizes to hundredths of an inch and the throat diameters to thousandths of an inch. The measurements on the throat diameters made by the author did not differ in any case more than 3

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

TABLE 1 PRINCIPAL CHARACTERISTICS OF PUMPS IN TORONTO PUMPING STATION

Service	Capacity in million gal per 24 hr	Suction-pipe diam, in.	Delivery-pipe diam, in.	Speed rpm	Pressures: Suction, ft	Pressures: Discharge, ft	Pressures: Net, ft	Stages in pump	Guar. over-all eff, per cent	Specific speed: Based on gal per min	Specific speed: Based on cfs
Wash-water pumps	6	14.91	11.93	730	36[a]	90	54	1	77.9	2365	112
	9	16.67	13.92	730	36[a]	90	54	1	79.2	2900	137
	12	18.29	15.93	730	36[a]	90	54	1	80.5	3350	158
Raw-water pumps	24	24.51	19.92	750	9[b]	73	82	1	84.5	3545	167
	30	26.58	21.94	750	9[b]	73	82	1	85.1	3970	187
	48	33.34	27.76	500	9[b]	73	82	1	86.0	3340	158
	60	38.68	33.92	500	9[b]	73	82	1	86.3	3740	176
Filtered-water pumps	6	15.05	11.91	750	36[a]	232	196	2	81.4	1556[c]	73[c]
	30	{24.97 24.85	{19.95 19.90	750	36[a]	306	270	2	86.0	2735[c]	128[c]

[a] Pressure head, ft.
[b] Suction lift, ft.
[c] Per stage.

parts in 17,100 from those made by the builders and, in most cases, the difference was within 0.001 in. This represents a high degree of accuracy when it is stated that the throat diameters varied from 9.53 in. to 27.2 in.

All those participating in the tests cooperated very earnestly in trying to secure precise results. To this end the contractor supplied mercury columns so that the pressures could be read directly in feet of mercury. In all but the raw-water pumps, there was sufficient suction pressure to allow the connection of the suction side of the pump to the top of the mercury column, while the delivery side was connected to the bottom of it. Thus, the mercury column read directly the net pressure head produced by the pump, automatically allowing for difference of elevation of the connections. For the raw-water pumps, only the discharge pressure could be taken on this column, the suction lift being read directly in feet of water. Great care was observed in clearing all piping of air and dirt and in adjusting the zeros of the measuring tapes; correction was made for the temperature of the mercury column and for the depression of the mercury in the pot corresponding to each pressure.

Four piezometer connections were provided for each pressure-measuring section, the openings being located in a plane surface normal to the axis of the nozzle and spaced 90 deg apart. Each connection consisted of a brass plug screwed into the nozzle wall having a $^1/_4$-in. hole bored through it, the inner edge of the hole being rounded to a radius of $^1/_{16}$ in. The plug was set so that its inner surface was flush with the inner surface of the nozzle wall and so that the axis of the hole was normal to the inner surface at the point of connection. The hole continued for over 2 diam without change of shape or direction. The inner surface of the pipe wall was scraped smooth and clean over an area approximately 6×12 in. immediately upstream from each piezometer opening, the 12-in. dimension being parallel to the axis of the pipe. Each opening was connected to a ring of brass piping and provided with a $^1/_4$-in. threaded connection on the ring midway between two of the piezometer plugs for use as a pressure-gage connection.

For the venturi meters it was decided to read the differential directly in feet of water, the maximum being about 100 in. The contractor furnished an inverted U tube for this purpose, the two water columns being brought to a convenient position by the use of compressed air. All piping to both this gage and to the pressure column was drawn copper tubing through which water was allowed to flow freely until all the air was driven out.

The gages were simple in construction, convenient for use, and not difficult to move to the different pumps. The contractor designed and built them all.

In computing the discharge, it was agreed by both parties to the contract that the venturi-meter coefficients recommended in the A.S.M.E. Fluid Meters Report should be used because, in all cases but one, the piping leading to the meter had a sufficient length of straight pipe. In one case, however, there was definite evidence that the recommended coefficient was too high, and a calibration of the meter by a volumetric measurement proved this to be the case. This matter will be discussed later.

Electric-input measurements were made by Weston instruments throughout, there being two current transformers, two potential transformers, two ammeters, one voltmeter, and a polyphase wattmeter. In order to check the instruments, they were set up in a standardizing laboratory in exactly the same way as they were to be used in the tests. In the laboratory, standard instruments were used to measure the line input corresponding to each indication of the various instruments. In this way all corrections for all instruments were included and the true value corresponding to each wattmeter indication was known. The ammeters and voltmeter were primarily used to check the power factor, which was kept as near 100 per cent as possible on all synchronous machines. The indications of the several meters required an exceedingly small correction.

Since the frequency of the system from which the electric power was drawn is maintained with great exactness at 25 cycles, it was not thought necessary to check the speeds of the synchronous motors, these being assumed as 500 rpm and 750 rpm. For the induction motors, the slip was calculated by means of a neon lamp and the speed of the motors was then taken as 750 rpm less the slip.

Recording of the observations presented a somewhat serious problem. In the first place, it was necessary to do all the testing between 11:00 p.m. and 6:00 a.m. and, for various reasons, it was not possible to adopt any regularity in the nights on which the tests were made, which made it difficult to arrange satisfactorily for observers. Again, it was necessary to carry out the work with the greatest precision possible and all personal errors had to be reduced to a minimum. In view of the well-known fact that in all commercial testing there is a continuous swinging of the electric-instrument needles, and also of the pressure gages and venturi differentials, it will be evident that observers might make considerable error in their readings. While the circumstances of night work, with little interference with the power lines of the electric system and with synchronous motors at fixed speed, tended to reduce these fluctuations to a minimum, yet they were always present and difficult to evaluate properly.

Photographic Recording of Instrument Readings

After a full consideration of all the circumstances, the author decided to take simultaneous photographs of the readings of all instruments during each test, the first result of which was to replace all the observers by a single photographer. Three cameras were used, a Rolleicoid with Zeiss $f/3.5$ lens for $2^1/_4$-in-square pictures with roll film on the electrical instruments, a camera with a film pack on the pressure measurements, and a camera with Zeiss Tessar $f/4.5$ lens for 9×12-cm plates on the venturi-meter differentials. To avoid parallax, the first camera was set directly over the wattmeter and one ammeter, Fig. 1, but in the

other ammeter and in the voltmeter there was a slight parallax error which could be corrected for, within the desired accuracy, by reading the position of the shadow of the needle as well as of the needle itself.

The position of the film-pack camera was adjusted so as to be approximately level with the top of the mercury column, and therefore parallax effect was insignificant. Furthermore, the measuring tape photographed with the mercury column was beside the latter and set out so that its face was opposite the center of the column, which naturally eliminated all errors. The greatest difficulty, although it gave little concern, occurred in photographing the differentials on the venturi meters as, in some tests, the differential was about 100 in., while in others only about 6 in. To complicate the situation further, the columns oscillated greatly, although the differential was fairly constant during a test; finally, water is not an easy fluid to photograph. The latter difficulty, however, was eliminated entirely by placing a white paper, with heavy black rules, at 45 deg to the vertical, just behind each column; the change in slope of the lines produced by the glass making it easy to locate the top of the water column. In photographing these columns, wooden scales graduated in divisions of $^1/_8$ in. were placed beside the columns so that the plane of the scales passed $^1/_8$ in. behind the center of the glass tubes. However, for the large differentials, a correction had to be made which will be discussed later.

With this camera, the operator ingeniously photographed two readings side by side on the same plate by covering up one half of the plate while exposing the other, a process which saved plates but rather increased the time required and at times was somewhat trying on the nerves. All exposures were made at 1/100 sec with incandescent lamps; the night work proving to be a help in that connection, since the illumination of the room was always the same, and the lighting of the apparatus was, therefore, easily kept uniform.

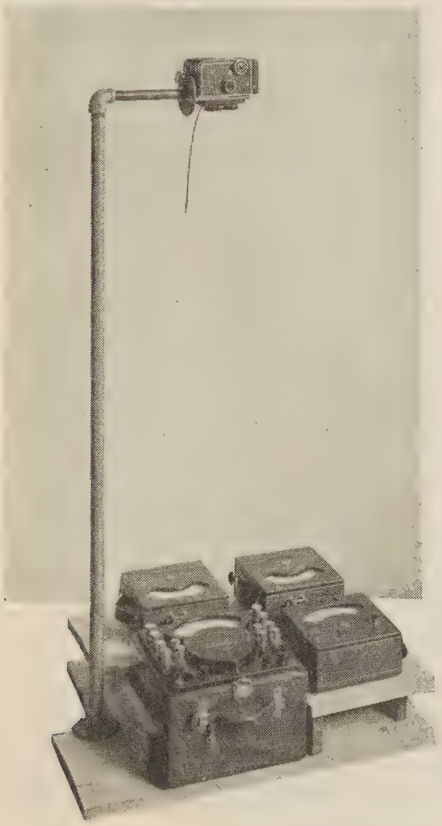

Fig. 1 Setup of Electrical Instruments With Camera

In making the exposures, an operator was placed at each camera, men being selected who could respond quickly to a signal. The shutter on each camera was set open, and each operator stood with his finger on the cable release; while a fourth man signaled the time. In each case two signals were given, the first 10 sec before the exposure, to get ready, and the second one to make the exposure. While the three cameras could have been operated simultaneously by relays, the method used produced such good results that it was never possible to detect more than one click when the exposures were made, and it would appear that all three exposures were made within $^1/_{10}$ sec of one another.

As the scales photographed with the columns were parallel to, but some distance away from them, some care was necessary in reading the records. Each photograph was set up under a microscope of suitable magnification, the degree being carefully chosen, since too high or too low a magnification made it impossible to read the water columns. The microscope was equipped with a movable table which could be adjusted in two directions at right angles to each other. With the glass column in the microscope set parallel to one direction of motion and with a cross hair in the microscope normal to this direction, the picture was moved till the cross hair was at the top of the column; the entire table was then traversed at right angles to the axis of the columns until the reading could be taken on the scale. Fortunately, a suitable microscope was available during these experiments, although for electrical instruments an ordinary microscope of rather higher power was used.

Readings of the photographs were taken by both the author and by H. Ulmann, pump designer, acting for the contractor, and any difference was at once adjusted, although in almost every case the two readings agreed.

Samples of the photographs are shown in Figs. 2, 3, and 4, Fig. 2 being from the mercury column, Fig. 3 from the venturi-meter tubes, and Fig. 4 from the electric meters. Unfortunately, they do not reproduce very clearly although the films were easy to read.

Accuracy of the Results

As far as observations go, the accuracy is beyond question, parallax having been avoided with the wattmeter and also in the mercury pressure column by adjusting the height of the camera in each test until it was almost exactly level with the surface of the mercury, and also by adjusting the plane of the measuring tape to suit the column. For the venturi-meter differential, some correction should be made, even though the plane in which the scales lay was nearly the same as that of the center of the tubes, because there was a bending of the light rays due to refraction, as illustrated in Fig. 5. This illustration shows a side elevation of the setup, and also an enlarged view of the tube and scale for the worst case during the runs.

In the enlarged diagram Fig. 5, the effect of refraction is shown, the refractive index for air being taken as $N_1 = 1$; for pyrex as $N_2 = 1.48$, a value determined experimentally for the tubes; and for water as $N_3 = 1.33$. The greatest value of θ_1 was 13° 40′ and, using the relation $N_1 \sin\theta_1 = N_2 \sin\theta_2 = N_3 \sin\theta_3$ the path of the ray has been drawn as indicated; the angle θ_2 is found to be 9° 25′, while $\theta_3 = 10° 13'$. The surface of the liquid was assumed to be spherical for this study and, by constructing the diagram on a large scale, the corrections c_1 and c_2 for the upper and lower columns in this extreme case were found to be $c_1 = 0.024$ in. and $c_2 = 0.026$ in. or, in other words, the apparent differential was 0.05 in. greater than the real quantity. This was corrected for in the large readings, although it only amounted to 0.05 per cent,

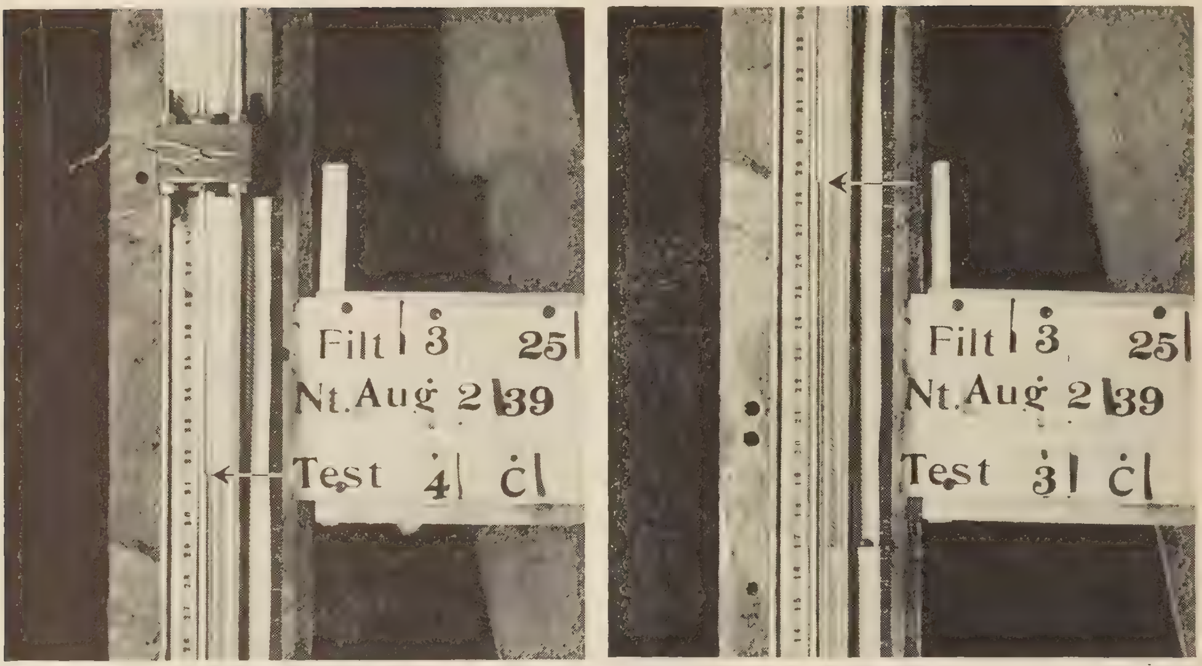

Fig. 2 Mercury Column for Pressure Head on Pump; Two Tests
(The tape is faintly shown at the left of the data card in each case.)

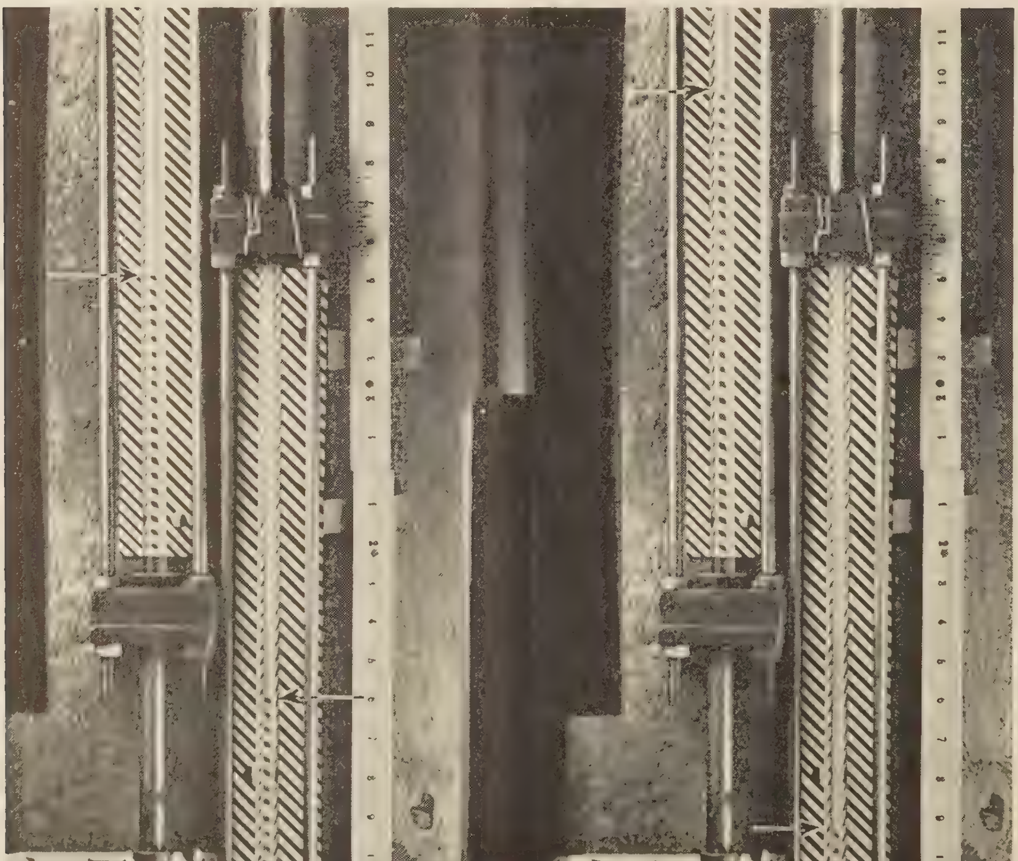

Fig. 3 Differential Gage on Venturi Meter; Two Tests

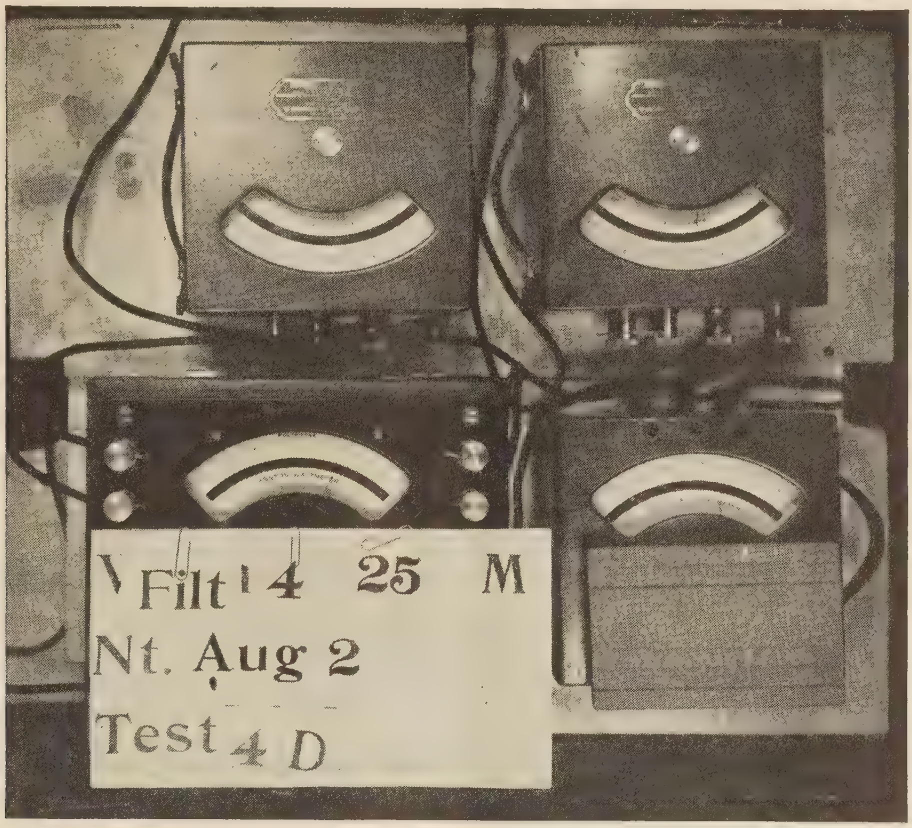

Fig. 4 Electrical Instruments Used in Tests
(The card states that the test was on filtered-water pump No. 4; capacity (25,000,000 imp gal) 30,000,000 gal per 24 hr; photograph taken on night of Aug. 2, and designated as test 4D. There is no blurring of the needles on the original film.)

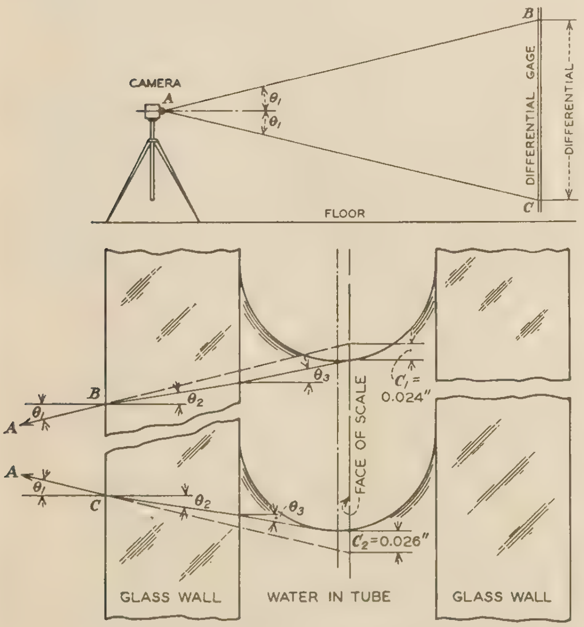

Fig. 5 Correction for Refraction

TABLE 2 EXTRACTS FROM TWO TYPICAL LOG SHEETS

Reference no.	Wattmeter reading	Pressure, in. mercury	Suction lift, in. water	Differential on venturi meter, in. water	Discharge valve
		Raw-water pump[a]			
1	0.738	63.5	26.0	102.4	wide open
	0.737	63.6	26.0	102.5	
	0.730	63.9	26.0	102.8	
	0.735	63.3	26.5	104.6	
2	0.732	67.5	28.5	91.8	partly closed
	0.740	67.6	27.5	93.2	
	0.738	68.0	26.0	90.7	
	0.730	67.1	25.5	92.2	
3	0.718	74.5	16.5	69.3	partly closed
	0.710	75.1	15.8	68.4	
	0.710	75.0	15.5	67.5	
	0.710	75.0	15.0	67.8	
		High-pressure pump[b]			
4	0.743	252.1	...	69.25	wide open
	0.738	250.2	...	68.75	
	0.748	252.0	...	68.72	
	0.740	253.3	...	68.20	
	0.738	251.1	...	68.50	
	0.740	251.7	...	69.25	
5	0.720	263.2	...	58.35	partly closed
	0.722	263.2	...	57.80	
	0.720	263.3	...	58.70	
	0.720	262.0	...	57.90	
	0.725	263.5	...	58.73	
	0.720	263.3	...	58.55	
6	0.652	283.8	...	36.65	partly closed
	0.655	283.5	...	37.05	
	0.652	283.8	...	36.90	
	0 654	283.7	...	37.30	

[a] Suction read separately.
[b] Net pressure directly taken.

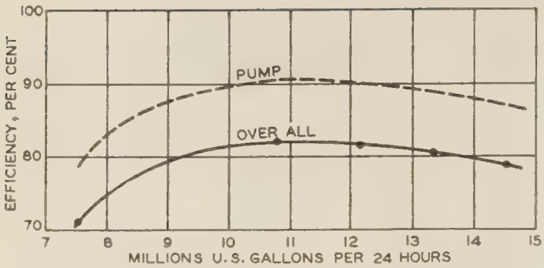

Fig. 6 Efficiency Curves for Wash-Water Pump
(Capacity 12,000,000 gal per 24 hr at 54-ft head and 730 rpm.)

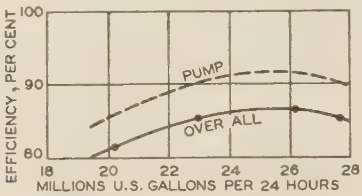

Fig. 7 Efficiency Curves for Raw-Water Pump
(Capacity 24,000,000 gal per 24 hr at 82-ft head and 750 rpm.)

while its effect on the calculated discharge is only 0.025 per cent. Of much greater importance is the weight of the air column balancing the differential; this was corrected for throughout.

The conditions under which these tests were made were as nearly perfect as one could possibly expect in commercial work, because the speeds were constant and the levels of the suction and discharge wells were uniform during tests, although there may have been some vibration in the valve disks, where these were used to regulate the discharge. The variations in the several quantities observed may be estimated from the brief extracts from typical record sheets, three entries for each test being given, Table 2.

Variations occurred in all quantities, irrespective of the time allowed for the pumps to attain uniform conditions. Where the discharge valve was wide open, as during best efficiency for the pumps, corresponding to Nos. 1 and 4, Table 2, the variations in the watts, pressure, and meter readings, were very small indeed, but with considerable throttling, as in Nos. 3 and 6, the variations were greater and the accuracy of the tests was less. In some of the tests, records were made over a fairly long time with results of the same nature. These facts are mentioned to show that variations of this kind do exist and are not due, as is often assumed, to the personal errors of the observers. Of course, the percentage variation in the computed discharge is only one half that in the observed differential.

The results of the tests may be of interest in view of the very high efficiencies obtained in the units; the curves for a typical unit of each group are given in Figs. 6, 7, 8, and 9. The guarantees made cover the over-all efficiencies and only these were measured in the tests. The motor efficiencies were measured at the builders' shops. When these results are applied to the over-

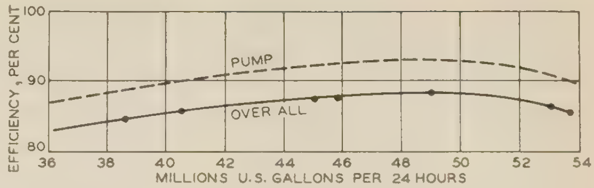

Fig. 8 Efficiency Curves for Raw-Water Pump
(Capacity 48,000,000 gal per 24 hr at 82-ft head and 500 rpm.)

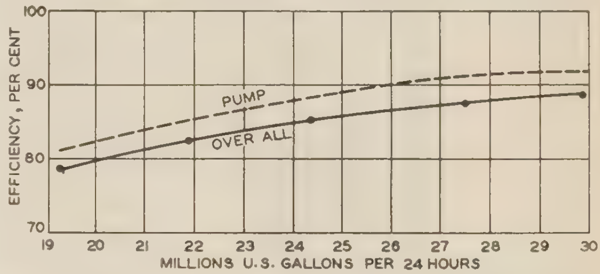

Fig. 9 Efficiency Curves for Filtered-Water Pump
(Capacity 30,000,000 gal per 24 hr at 270-ft head, two stages, and 750 rpm.

all efficiencies determined by the pump tests, the pump efficiencies are obtained and these are also shown in the illustrations.

Model Tests

Models were made of the various pumps by the Dominion Engineering Works, Montreal and, through the courtesy of that company, the author has been permitted to give some of the results of the tests on the models made by the contractors at their shops. These results are for models of the pumps, corresponding to the efficiency curves of Figs. 6, 7, 8, and 9, i.e., for the 12,000,000-gal wash-water pump, the 24,000,000- and 48,000,000-gal raw-water pumps, and the 30,000,000-gal filtered-water pump. The models were, of course, made with the same specific speeds as their respective prototypes and in Table 3, data on the prototype and model are given, together with the efficiency obtained in each case; the specific speeds are based on discharges in cubic feet per second.

The laws governing the construction of centrifugal-pump models will be briefly reviewed. The specific speed of the model and prototype must, of course, be the same, and attention must be given to the smoothness of finish of the model. The variables in the relationship between the model and the prototype may be made by the method of dimensionless numbers.

Let Q, D, N, H be the discharge, impeller diameter, speed, and head for the prototype, while ρ, μ, g are the density, viscosity, and gravity attraction on the water pumped by it; further, let q, d, n, h, ρ_m, μ_m, g_m represent the corresponding quantities for the model. Then evidently it is possible to write

$$Q = \phi(D, N, H, \rho, \mu, g)$$

which, according to the theory of dimensionless numbers, may be written

$$Q = \phi(D^a \times N^b \times H^c \times \rho^e \times \mu^f \times g^k)$$

TABLE 3 PUMP AND MODEL DATA

Prototype: Capacity, million gal per 24 hr	Prototype: Q cfs	Prototype: H ft	Prototype: N rpm	Prototype: Pump efficiency E	Specific speed, Ns	Model: q cfs	Model: h ft	Model: n rpm	Model: Pump efficiency e
12	18.54	54	730	0.902	158	9.4	67.0	1200	0.900
24	37.08	82	750	0.915	167	11.1	92.8	1500	0.896
48	74.16	82	500	0.930	158	9.8	68.5	1200	0.896
30	46.35	135 per stage	750	0.918	128	10.2	91.8 per stage	1200	0.907

TABLE 4 EFFICIENCY RELATION BETWEEN MODEL AND PROTOTYPE

Pump capacity, million gal per 24 hr	Speeds, rpm		Head, ft		Calculated $\frac{D}{d}$	$\frac{H}{h}$	Efficiency		
	Pump	Model	Pump	Model			Measured on model e	Calculated for pump E	Measured on pump
12	730	1200	54	67.0	1.48	0.81	0.900	0.908	0.902
24	750	1500	82	92.8	1.88	0.88	0.896	0.910	0.915
48	500	1200	82	68.5	2.63	1.20	0.896	0.920	0.930
30	750	1200	135	91.8	1.94	1.47	0.907	0.924	0.918

Substituting dimensions and using M, L, T for mass, length, and time

$$L^3T^{-1} \equiv L^aT^{-b}L^cM^eL^{-3e}M^fL^{-f}T^{-f}L^kT^{-2k}$$

Collecting coefficients

$$3 = a + c - 3e + e + k \qquad \text{Hence } a = 3 - c + 2e - k$$
$$1 = b - e + 2k \qquad b = 1 + e - 2k$$
$$0 = e + f \qquad f = -e$$

Therefore

$$Q = \phi(D^{3-c+2e-k} \times N^{1+e-2k} \times H^e \times \rho^e \times \mu^{-e} \times g^k)$$

or

$$\frac{Q}{ND^3} = \phi_1\left(\frac{H}{D} \times \frac{g}{DN^2} \times \frac{ND^2}{\nu}\right)$$

where ν is the kinematic viscosity.

The kinematic viscosity of the water used is the same for the model and prototype, and experience indicates that the term $\frac{ND^2}{\nu}$ has little effect. Theory of the pump further suggests that the first and second terms are connected as a product and, further, $Q/(ND^3)$ is proportional to the specific speed and must be the same in model and prototype. Therefore, it follows that

$$\frac{H}{D^2N^2} = \frac{h}{d^2n^2} \quad \text{or} \quad \frac{D}{d} = \frac{n}{N}\sqrt{\frac{H}{h}}$$

Since the pump and turbine are exactly similar, it seems reasonable to apply Moody's formula for the efficiency relation between model and prototype, and this is

$$\frac{1-E}{1-e} = \left(\frac{d}{D}\right)^{1/4}\left(\frac{h}{H}\right)^{1/10}$$

Applying these principles to the four pumps considered gives the results in Table 4.

For the 12,000,000-gal, 24,000,000-gal, and 30,000,000-gal pumps, the calculated efficiency agreed very well with the observed value, but the 48,000,000-gal pump actually gave 1 per cent higher efficiency than the model indicated. As this efficiency was unusually high, the tests were repeated and the venturi meter was calibrated, but there appears to be no error in the figures given.

Venturi-Meter Coefficient

Owing to the very high efficiency indicated on the 48,000,000-gal pump, the author decided to eliminate all possibilty of error by calibrating the venturi meter against definite displacement in one of the reservoirs at the station. Before beginning the calibration, and after finishing it, very careful observations were made on leakage, and these showed the reservoir to be perfectly tight. The volume of the reservoir was computed from measurements and the test was continued for 5 hr with as nearly constant differential on the meter as was possible to keep by a control valve which was constantly watched. The depth of water delivered to the reservoir was 12.88 ft and, as the elevations could be measured to about 0.06 in., there should be no error in that measurement; the differentials on the meter were photographed at 5-min intervals during the entire run. The results were as follows:

Size of meter........................	27.2 × 41.78 in.
Coefficient obtained on this test and used........	0.9545
Coefficient recommended by Fluid Meters Committee....................................	0.9876

Acknowledgment

These pumps have set a very high standard for units of this size and duty, and reflect great credit on the Dominion Engineering Works, Montreal, and particularly on their chief engineer, Mr. H. S. Van Patter, and on their pump designer, Mr. H. Ulmann.

The author's thanks are due to Mr. R. C. Harris, Commissioner of Works, Toronto, for permission to publish these results. The rigid specifications drawn up under his direction are undoubtedly partly responsible for the splendid showing. Thanks are gladly tendered to Mr. A. U. Sanderson and Mr. L. F. Allen, engineers of the Works Department, for their cooperation in all the work; particularly Mr. Allen who was present throughout every test and was responsible for the operation of the pumps. Mr. L. E. Jones assisted with photography and showed great skill in the use of the cameras and in the lighting, and Mr. Ulmann who was present during all of the tests showed the author every courtesy in a rather difficult task.

Discussion

L. F. Allan.[2] The author gives a comprehensive description of the methods used in testing the over-all efficiencies of the pumps recently installed at Victoria Park, Toronto. The arrangement of the measuring sections which, in general, follows the recommendations of this Society was described in the specifications on which the contract for the supply and installation of the pumps was based. A conference of all parties interested in the tests decided that a high degree of accuracy was required and that water manometers for the venturi meters, mercury gages for measuring pressures, and cameras for taking the readings should be used.

The advantages of photography for recording the indications of the gages and instruments are obvious, and the practicability of the method has been demonstrated. Its chief advantage is the avoidance of the inherent human errors involved where a number of observers are employed. With cameras, the only human element entering into the recording of the indications is the operation of the camera shutters, and this can be done quite accurately. Other advantages are the reduction in the number of persons required for the tests and the securing of a permanent photographic record of all indications.

Although the design and arrangement of the apparatus received earnest consideration at the outset, the technique developed gradually as defects and difficulties appeared. For example, the first photographs of the water manometer showed

[2] Department of Works, Water Supply Section, Toronto, Canada.

insufficient contrast between the part of the glass tube containing water and that containing air, so that the position of the top of the water columns was not easily readable. To increase the contrast, it was suggested that dye be placed in the water but the idea was discarded because it was thought that, following a fluctuation, the dye might stick to the glass tube thus blurring the image. A white card with sloping black lines was placed as a background behind each glass tube. With this arrangement, due to lens effect, there is in the image an abrupt change in the slope of the lines at the water surface.

Another difficulty which presented itself at the outset was the glare from the lights reflected from the glass tubes, which made the negatives less easy to read. To improve this condition, the lights were counterbalanced on a vertical support so that by raising or lowering them the reflections did not occur at the water surface. The glare was later entirely eliminated by arranging a shield in the form of a black cloth tape alongside each glass tube of the manometer so that light did not shine on the glass tube but did illuminate the background, whence it was reflected through the tube to the camera. The same device was applied to the mercury tube. In the displacement test for checking the meter coefficient, an electric clock was included in the photographs; it was found that the clock should be placed near the center line of the lens in order to avoid the effects of parallax. The clock should previously be checked to insure that the minute hand takes the correct position at all points on the dial and it should not be entirely relied upon lest it stop due to power failure. Stop watches were used to obtain elapsed time of the test, etc.

In connection with the mercury gage, there was only one column to photograph and it was possible to mount the camera on a sliding carriage on a pipe standard so that the camera was always positioned horizontally opposite the surface of the mercury column. Any variation of the location of the image from the center of the negative was due to fluctuation of the pressure.

This method is more accurate and convenient than other methods, yet its accuracy and speed can be further increased by certain refinements not difficult to arrange. For instance, all cameras could be made to operate at the pressing of a single button which would have the advantage in that one person could set and control all the apparatus connected with taking the pictures. The parallax due to the wide angle made with the camera by the two water columns at the manometer could be avoided by using one camera for each column. These cameras could be synchronized mechanically to operate at exactly the same time. The light for illuminating the mercury column could be projected through a slot behind the glass tube and would reach the camera lens only in the portion above the mercury, thus providing maximum contrast. A similar arrangement with a suitable background could be used for the water manometer.

An alternative method which would perhaps yield more information would include the use of motion-picture cameras, and an electric clock in the field of each, all properly synchronized to produce simultaneous exposures correctly identified.

Practically any type of camera having a shutter speed of 1/100 sec can be used. In these tests, four types were employed, i.e., a plate camera, a reflex camera, a film-pack camera, and a 35-mm miniature camera; each did its work well. The reflex camera has a special advantage in that the focus can be checked at all times. The value of this was illustrated in one case when the setting of one camera was disturbed and the pictures were blurred so that the entire test had to be discarded. The photographic method has proved so satisfactory that one would be loath to revert to the use of observers. Indeed, under the unsteady conditions encountered, it is doubtful if observers would be satisfactory.

The endeavor to insure a high degree of accuracy, by using mercury pressure gages, water manometers, and cameras, was extended to the erection of the apparatus and the taking of essential data necessary for the calculations. Measurements were made independently by more than one operator and the author invariably checked each one to eliminate any possibility of error.

Charles Babb.[3] The author's tests indicate that the prototype in this instance showed an increase in efficiency over model tests which follows very closely the Moody formula for efficiency relation between model and prototype. Three manufacturers of pumps have discovered that the expected increase in efficiency did not materialize in the case of the pumps for the Colorado River Aqueduct.

Contractors' models, which were approximately one fifth the size of the prototype were built and tested in the hydraulic laboratory at the California Institute of Technology. All water passages of the model had to be in exact ratio to those of the prototype. They were very carefully constructed and were given the best finish possible, in order to obtain high efficiencies. They were carefully tested with the finest of precision instruments, both in the manufacturer's laboratory and at the institute.

The prototypes were manufactured with the same care as were the models, particularly with respect to finish, polish, and paint. They were tested in the field with the same care as were the models, and with the best of calibrated precision instruments.

The Allen salt-velocity method was used to determine the capacity. Preliminary results gave lower pump efficiencies than were expected. The penstocks were extremely smooth and were at quite an incline. The Allen technicians installed turbulators ahead of the salt pop valves and more consistent results were then obtained. A 10-acre reservoir in the immediate foreground of the Iron Mountain plant was carefully surveyed and was used as a volumetric calibration. Tests were taken to determine the inflow and outflow corrections which should be applied. Tests were performed at night to minimize the evaporation loss. The results of the calibration showed that the Allen-method values were from 0.1 to 0.6 per cent higher than the volumetric measurement, varying with the number of pumps in operation.

The final results of both tests on the Allis-Chalmers pumps gave 92.5 per cent maximum efficiency for the model and 91.3 per cent average efficiency for the prototypes. Similar results were obtained on all the different manufacturers' pumps. We are, therefore, of the opinion that not too much faith should be placed in obtaining increased efficiency for prototype over model.

With regard to venturi-meter coefficient, we had some experience with the meter results changing in a very short time at the laboratory at the institute. The meter coefficient changed from 0.3 to 0.4 per cent between calibrations. It is the writer's opinion that all venturi meters should be calibrated, if possible.

R. L. Daugherty.[4] The title of the paper appears to be somewhat misleading, since there is nothing new in the methods of measurement as described. If it were designated as an "Improved Technique for Field Tests," that title would be more appropriate. The procedure used by the author is actually a step toward bridging the gap which usually exists between the accuracy of field tests and laboratory tests. Thus, the quantity of water is measured by venturi meters, which is common laboratory practice, although not so often encountered in actual installations of large pumps. However, only one of these meters was calibrated, while the remainder employed the coefficients

[3] Design Engineer, Centrifugal Pump Division, Hydraulic Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis.

[4] Professor of Mechanical Engineering, California Institute of Technology, Pasadena, Calif. Fellow A.S.M.E.

recommended by the A.S.M.E. Fluid Meters Committee. The reason given for calibrating this one meter was that the recommended coefficient gave a pump efficiency of 0.965 per cent. The calibration of the meter yielded a lower coefficient than the recommended value and thus reduced the pump efficiency to 0.93 per cent. The question therefore arises why all of the venturi meters were not then calibrated? Although all of the other pump efficiencies reported were not unreasonable values, they might also have been lowered 2 or 3 per cent by the use of lower meter coefficients. Aside from the fact that the recommended coefficient for this one meter gave too high a pump efficiency, was there any other reason why it would appear to require calibration more than the others used?

The employment of manometers for the differential gages on the venturi meters and mercury manometers for the measurement of pressure heads on the pumps is common laboratory practice. This is also found in occasional field tests, though of course it is not so common there as in laboratories.

The use of calibrated electrical instruments for the measurement of input to the electric motors is common practice for good field tests. Likewise, the acceptance of motor efficiencies as supplied by the motor manufacturers is common, but would not be considered good laboratory technique.

The use of cameras to photograph the readings of the various instruments used is good practice and has been much utilized in laboratory work, but not very frequently in field tests.

It is stated that the frequency of the electric power used is maintained at great exactness at 25 cycles and, therefore, the pump speeds were not measured in the cases of the synchronous motors and only the slip measured for the induction motors. It would be interesting to know what evidence is available as to the exactness of the frequency of the system at Toronto. In Pasadena, we find sudden deviations of the order of $^1/_4$ per cent, which would produce $^3/_4$ per cent variation in horsepower. While the local frequency is uniform over a period of time, the instantaneous variations would interfere seriously with precise measurements in our hydraulic-machinery laboratory.

A most interesting part of this paper is the comparison of the efficiencies of the pumps tested by the author and of the model pumps tested by the manufacturers. The differences between model and prototype are just about what one should expect. Also the agreement of the prototype-test efficiencies with the values computed from the models by the Moody formula is very gratifying.

L. E. Jones.[5] Although references to the use of photography in scientific and technical work are legion, there does not appear to be much available in the literature on the particular method employed by the author. Therefore, it would appear to be desirable to set forth here a brief account of the photographic details involved. The author remarks on the high accuracy obtained in the observations, but it must be emphasized that this result is brought about by design rather than by chance. In the following discussion there will be considered the various factors, peculiar to the particular methods of observation, which affect the accuracy of the readings, with indications as to means of computing the necessary corrections. This analytical treatment is given principally to show the prevention rather than the cure, as it is generally much safer and certainly much more convenient to minimize corrections beforehand by proper arrangement of the equipment. These corrections, of course, are not the only ones required in working up the results, but only those which are caused primarily by the use of photography will be considered.

[5] Instructor, Department of Applied Physics, University of Toronto, Toronto, Canada.

For detailed descriptions of photographic theory and technique, the reader is referred to standard works on the subject.[6]

Preliminary Optical Calculations

Before any testing procedure is attempted, it is essential that the photographic apparatus be arranged to give the best results and, while there is no improvement on actual trials, much time can be saved by preliminary calculations. For this purpose, certain optical equations are presented herewith, with indications as to their use.

The following nomenclature will be used:

f = focal length of camera lens[7]
U = object distance = distance of object from (entrance node of) lens
V = image distance = distance of image from (exit node of) lens
d = distance from object to image (neglecting internodal distance), all measured parallel to optical axis of lens
y = given dimension of image
Y = corresponding dimension of object; both measured perpendicular to optical axis of lens
M = linear magnification or scale of image[8]
$= \dfrac{\text{Length of image (measured perpendicular to optical axis)}}{\text{Length of corresponding part of object}}$

According to the "law of lenses"

$$\frac{1}{f} = \frac{1}{V} + \frac{1}{U} \quad [1]$$

whence

$$f = \frac{UV}{U + V}; \quad U = \frac{Vf}{V - f}; \quad V = \frac{Uf}{U - f} \quad [2]$$

$$M = \frac{y}{Y} = \frac{V}{U} = \frac{f}{U - f} = \frac{V}{f} - 1 \quad [3]$$

$$f = \frac{Md}{(M + 1)^2} \quad [4]$$

$$U = \frac{f(M + 1)}{M} \quad [5]$$

The emulsion resolving power governs the fineness of detail which can be distinguished in the photograph, and the magnification M must be chosen to suit the precision required in the linear measurements. As a general rule, the smallest quantity to be measured on the photograph should not be less than $^1/_{200}$ in., and definitely should be greater if at all possible, as it will make the interpretation much more satisfactory. If Y is the maximum length of object (for example, the maximum venturi differential), the corresponding size of image is $y = MY$, and the size of plate or film should be chosen to give an adequate margin of safety. Once this size and the corresponding lens has been selected, the location of the camera may be computed from Equation [5].

[6] "The Photographic Process," by J. E. Mack and M. J. Martin, McGraw-Hill Book Co., Inc., New York, N. Y., 1939.
"Handbook of Photography," by K. Henney and B. Dudley, Whittlesey House, New York, N. Y., 1939.
"Photography, Theory and Practice," by L. P. Clerc, Pitman Publishing Company, New York, N. Y., 1937.

[7] On most lenses the value of the focal length will be found engraved on the mount. This is the nominal value, which may differ slightly from the actual value. Although not generally required, the latter may be conveniently determined by measuring M and d and solving Equation [4], provided d is large enough to make the internodal distance negligible. Other methods of determining f are given in the standard works.[6]

[8] For most classes of work, the value of M will be less than unity.

With these preliminary data, it is possible to compute the time of exposure required to "stop" adequately the fluctuation of the gage column or meter needle. If x is the estimated velocity of fluctuation in inches per second, M the magnification, and T the time of exposure in seconds, the image velocity will then be Mx inches per second, and the image movement during exposure MxT inches. Due to the limited resolving power of the eye, the blur resulting from this image movement will be satisfactorily "sharp," provided it does not exceed a certain size (for example, $^1/_{500}$ in.) and, by equating this value to the quantity MxT, the time of exposure may be determined. Other considerations, such as floor or tripod vibration, may necessitate shortening this time. With this determined, adequate illumination must be provided to give the required exposure on the photographic emulsion. The lens aperture should be fairly small (around $f/8$) to utilize the best characteristics of the lens and to give some depth of field.

Interpretation of the Photographic Records

The principal reason for correcting the photographic readings lies in the fact that the camera draws by central projection (perspective) rather than by orthographic projection, which is the condition automatically assumed by the visual observer. The two types of measurements made in the pump tests are those involving fluid manometers and electrical instruments, and these will be dealt with separately. It is assumed, of course, that the camera objective is of good quality (i.e., corrected for lens aberrations and distortion) and adjusted in proper focus.

1 *Fluid Manometers.* The remarks here are concerned primarily with measurement of the venturi differential, as this involved the greater difficulty. The arrangement of gage tubes and scale is shown in Fig. 3 of the paper and schematically in Fig. 10*A* of this discussion, and the required differential head is the vertical distance between similar points on the two menisci. This distance is taken between the imaginary horizontals *ab*, *cd*, Fig.

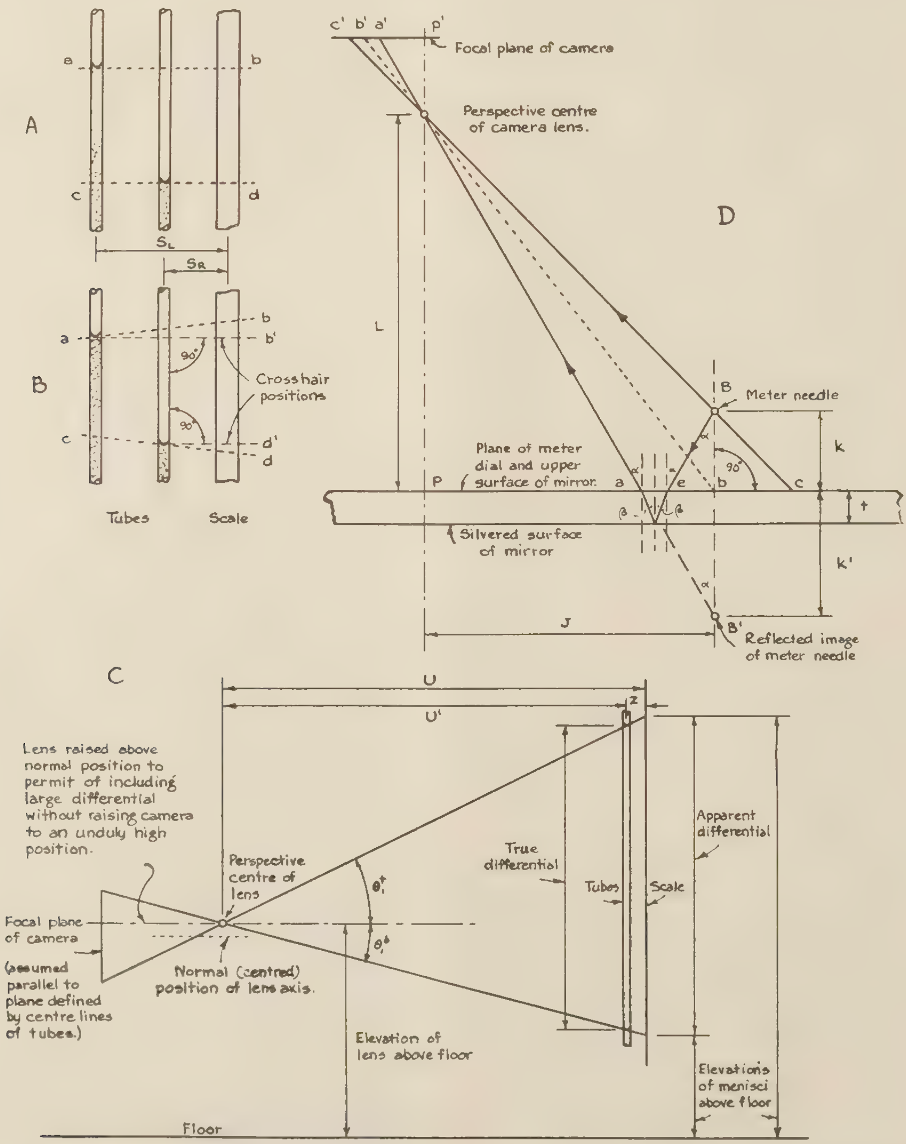

Fig. 10 Schematic Arrangement of Photographic Apparatus and Focal Details

10A, tangent respectively to the two menisci, and is read by means of a traveling microscope, as explained by the author. If this method of reading is to be valid, however, the "rectangle" *abdc* must be reproduced as such in the camera, which requires (1) that the scale be located in the same plane as the center lines of the tubes, and (2) that the camera back be parallel to this plane. Departure from either or both of these conditions results in a distortion of the form displayed in Fig. 10B, with obvious inaccuracies in the results. (If the conditions producing this distortion were in the opposite direction, then side *bd*, Fig. 10B, would be less than *ac*.) It is generally much simpler to treat these conditions separately as follows:

(*a*) Focal plane of camera parallel to plane of manometer tubes, but scale either in front of or behind this plane. This is the condition depicted in Fig. 10C, and gives rise to the obvious relation:

$$\frac{\text{True differential}}{\text{Apparent differential (as read from photograph)}} = \frac{U'}{U} = 1 \pm \frac{z}{U}$$

where, U' = distance from lens to tube axes
U = distance from lens to scale (which may be greater or less than U')
z = axial distance between tube axes and scale

If the venturi differential is very large, it is often impossible or unwise to set the camera up midway between the two extremes of the differential, and in such circumstances it is necessary to raise the lens with respect to the camera back, as shown in Fig. 10C. Tilting of the entire camera is of course undesirable. If needed for the refractive correction mentioned under section (*c*), the partial angles of view, θ_1^t (top) and θ_1^b (bottom), may be computed from the respective elevations of the lens and the two menisci, easily obtained from measurements above floor level.

(*b*) Scale in same plane as manometer tubes, but camera back not parallel to this plane. While simple enough, by means of a spirit level, to make the camera back vertical, it is not so easy to eliminate swing in the horizontal direction (i.e., in azimuth). If ψ is the amount of this swing, and θ_1 (θ_1^t or θ_1^b) the angular elevation of the ray passing through the meniscus, the amount of the discrepancy resulting due to these conditions is given, in magnitude only, by the expression $S \sin \psi \tan \theta_1$,[9] where S is the transverse distance between gage tube and scale, Fig. 10A. This discrepancy corresponds to distance *bb'* or *dd'* (*aa'* or *cc'* in the opposite condition) shown in Fig. 10B. Representative values are given in Table 5 of this discussion for S = 1 in. and, since the discrepancies for upper and lower readings are additive, the error may easily become appreciable.

The values of Table 5 are primarily of academic interest, as it is usually difficult to determine the amount of swing ψ. The simplest and best method of correction is to place the scale as close as possible to the manometer tubes (preferably between them), and to exercise care in aligning and leveling the camera. Otherwise two scales should be used, one on either side of the manometer and with their zeros accurately adjusted to the same level. Thus, whatever the resultant shape of the original rectangle, the horizontals *ab* and *cd* will be always defined, respectively, by like graduations on the two scales; this may, of course, involve some additional labor in reading. A word of caution should be given regarding linear measurements taken from photographs, as the linear magnification will be nonuniform from point to point except under strict conditions of parallelism. Hence, the necessity, apart from the convenience, of referring the measurements to an actual scale incorporated with the manometer.

(*c*) Even if the errors considered in sections (*a*) and (*b*) are eliminated, there still remain the corrections due to refraction. These have already been discussed by the author, and it will suffice here to indicate how the values may be obtained analytically. The following equations assume that the scale is in the plane of the tube axes, and that the camera back is parallel to this plane. In computing these corrections, the value of θ_1 (θ_1^t or θ_1^b) is obtained as suggested under section (*a*), and the other angles follow from Snell's law of refraction:

$$N_1 \sin \theta_1 = N_2 \sin \theta_2 = N_3 \sin \theta_3, \text{ etc.}$$

where N is the absolute index of refraction and θ is the angle between the ray and the normal to the refracting surface; subscripts refer to the different media. The corrections to the meniscus readings are always negative and are as follows:

For top meniscus

$$c^t = W[\tan \theta_1^t - \tan \theta_2^t] + \frac{B}{2}[\tan \theta_1^t - (1 + \sin \theta_3^t) \tan \theta_3^t + \text{vers } \theta_3^t]$$

For bottom meniscus

$$c^b = W[\tan \theta_1^b - \tan \theta_2^b] + \frac{B}{2}[\tan \theta_1^b - (1 - \sin \theta_3^b) \tan \theta_3^b - \text{vers } \theta_3^b]$$

where W = wall thickness and B = bore of tube. Table 6 of this discussion shows representative values for assigned values of θ_1. The values given considerably exceed those encountered during the tests, and are included merely to demonstrate the trend of the correction. Items 3, 3*a*, and 3*b* of Table 6 show the effect of unequal angles above and below the optical axis, while item 4 shows what might be expected if a "wide-angle" lens were used on the camera to permit photographs to be taken in cramped quarters.

Such corrections as those indicated should be quite satisfactory if actual conditions were the same as those assumed, but this is seldom likely to be the case, as irregularities in the glass tubing or slight deviations from the vertical would alter considerably the optical characteristics. By far the safest procedure is to keep the angular subtense as small as possible by proper arrangement of the camera; any correction which may arise will then be small, and the calculated value not likely to depart much from the proper value. Although the equations appear rather formidable, a few values will suffice to plot curves (corrections against θ_1) adequate to cover any given series of tests.

With the mercury head gage, the recording is usually much

TABLE 5 ERRORS IN RECORDED POSITION OF ONE MENISCUS DUE TO NONPARALLELISM OF MANOMETER AND CAMERA BACK (BOTH VERTICAL)

(Transverse distance between tube and scale = 1 in.)

Elevation of ray θ_1	Horizontal swing ψ: 5 deg	10 deg	15 deg
deg	in.	in.	in.
5	0.008	0.015	0.023
10	0.015	0.031	0.046
15	0.023	0.046	0.069

TABLE 6 REFRACTIVE CORRECTIONS IN VENTURI MANOMETER

(Scale assumed to be in plane of tube axes; camera back parallel to plane of manometer)

Item	Angle subtended by differential: Total, deg	θ_1^t (top), deg	θ_1^b (bottom), deg	Corrections: Top meniscus, in.	Bottom meniscus, in.	Total, in.
1	10	5	5	0.008	0.008	0.016
2	30	15	15	0.024	0.028	0.052
3	60	30	30	0.055	0.071	0.126
3*a*	60	15	45	0.024	0.151	0.175
3*b*	60	45	15	0.115	0.028	0.143
4	90	45	45	0.115	0.151	0.266

NOTE: Wall of tube = 0.21 in.; bore of tube = 0.20 in. Refractive indexes: Air 1.00; Pyrex glass 1.48; water 1.33.

[9] This assumes that the lens axis is normal to the camera back, and directed centrally toward the subject.

simpler, as the camera may be placed close enough to give a convenient value of magnification, and raised or lowered to reduce the effects of obliquity. The preceding remarks on manometer corrections apply equally well, of course, with the exception that there is only one meniscus involved and, thus, corrections (*a*) and (*b*) may be considered together. With suitable technique, it should be possible to eliminate the need of corrections almost entirely, but it must always be borne in mind that, for any given tolerance in the final result, finer readings must be made with mercury than with water in the manometer.

For locating the mercury meniscus, it is of considerable value to have water above the mercury column, with a diagonal pattern behind as previously explained. This water is always present when the gage is used as a differential meter, but when used for measuring discharge pressure only, the water may need to be added separately to the top of the column. In the latter case, the water meniscus gives a convenient check on the mercury reading, and may be invaluable if the mercury meniscus should be obscured by a manometer joint.

2 *Electrical Instruments.* When the instruments are used in the horizontal position, the camera must be supported with the lens axis vertical and, to permit satisfactory definition, the meters should be arranged so that their dials lie at the same elevation, Fig. 1 of the paper. The only observational correction which appears to be necessary is that due to obliquity of the rays and, as most meters are provided with a plane mirror for the elimination of visual parallax, this happily provides a simple means for determining the proper reading. Fig. 10*D* of this discussion is drawn vertically through the perspective center of the camera lens normal to both the meter needle and the plane of its dial, and illustrates in exaggerated fashion the conditions encountered when photographing a single meter. Due to the obliquity, the actual needle B appears to be "opposite" point c on the dial, while its reflection B'[10] appears to be "opposite" point a on the dial; the photographed images of these points lie, respectively, at c' and a'. The true reading of the needle occurs directly below it at the point b, and should be recorded in the photograph at b'.

Using the notation of the figure, we may show by similar triangles that

$$\frac{ab}{Pb} = \frac{k'}{L + k'} \text{ and } \frac{bc}{Pb} = \frac{k}{L - k}, \text{ whence } \frac{ab}{bc} = \frac{k'}{k} \times \frac{L - k}{L + k'}$$

If L, the height of the camera lens above the meter dial, is great with respect to the distances k and k', and if the focal plane of the camera is parallel to the meter dial (both of which are generally the case), then $\frac{a'b'}{b'c'} = \frac{ab}{bc} = \frac{k'}{k}$, which, when evaluated, permits of interpolating the true dial reading from the two recorded approximate readings. As a first and usually adequate approximation, k' may be assumed equal to k, and it thus suffices to take the arithmetic mean of the dial readings represented by a' and c'. If necessary to refine the process, we may proceed thus from the figure: $\frac{k'}{k} = \frac{ab}{eb} = \frac{ae + eb}{eb} = 1 + \frac{2t \tan \beta}{k \tan \alpha}$. The angles α and β are connected by Snell's law of refraction and, since the angles are generally small, we may replace the ratio $\tan \alpha/\tan \beta$ by $\sin \alpha/\sin \beta = N$, the index of refraction of the glass of the mirror, with respect to air. To sufficient accuracy then, the "interpolation factor" is $\frac{k'}{k} = 1 + \frac{2t}{kN}$, which is constant. Actual measurement of the quantities t, k, and N for any particular meter is obviously not desirable nor is it necessary, for the factor may be obtained by the simple means of reversing the procedure mentioned; in any event, the nature of the factor does not require great accuracy in its determination. That is, with the actual meter in use, values represented by a, b, and c are read and the calculations reversed to determine the interpolation factor; readings a and c are made from the same viewpoint to one side of normal and not too close to the dial, and reading b is made in the usual normal position. Care must be taken to apply the interpolation factor in the proper way, which gives the ratio $\frac{a'b'}{b'c'} = \frac{ab}{bc}$ greater than 1; in Fig. 10*D* this ratio is somewhat less than 1 due to the exaggeration involved.

Since the cover glass protecting the meter has parallel faces, it has no appreciable effect on the optical conditions just discussed.

If the meter dial has no mirror, then only one needle reading will be recorded, namely, that at c. The correction is therefore

$$bc = Pb \times \frac{k}{L - k} = \frac{Jk}{L - k} = \frac{Jk}{L}, \text{ approximately.}$$

If there is any doubt as to the direction of obliquity (and thus as to the sign of this correction), it may usually be determined from the perspective of the photograph.

The parallax for any meter can be completely eliminated by placing the pivot of the needle directly below the camera lens. However, since it is only the component of obliquity tangent to the needle which has any effect on the reading, it is possible to minimize parallax for more than one meter, if the lens is vertically above the line of the average needle position of each instrument.

As with the venturi gage, it is desirable to reduce the angular subtense by crowding the meters close together and using a camera lens of not too short a focal length.

Photographic Technique

The fact that the photographs referred to in the paper were made with standard camera equipment should counteract any criticism that this method of observation belongs essentially to the laboratory. It is true that something more than "snapshot" technique is required, but photographs such as those described can be made, under any reasonable circumstances, by an enthusiastic technician and intelligent, though not necessarily technical, assistants, even if necessary to improvise a temporary darkroom on the job. It is not necessary or desirable to make prints from the negatives (as some of the detail may be lost in the printing process) and observations may thus be made as soon as the negatives are dry. With suitable arrangements beforehand, little more than a strong magnifying glass should be required to assist in making the readings. The following remarks will be devoted to a brief consideration of the practical aspects of the photography.

Since the number of photographic exposures to be made in any extensive series of tests will be considerable, it is essential to use cameras which provide for minimum expense and easiest handling of the negative material during exposure and subsequent processing. Ease of operation during the actual tests is extremely important, particularly with operators unaccustomed to cameras, in order that the operator's mind may be free to concentrate on the business of making the exposures at the proper times. Any involved process, mental or manual, may result in missed "shots," which may often spoil a whole set and waste much time and material. Due to their cheapness and great

[10] Not its "shadow," as suggested by the author. In the tests, the parallax in two of the instruments was eliminated by suitable placement of the camera, so that needle reflections were observed only for the other two instruments, viz., those shown on the right of Fig. 4 of the paper. Unfortunately, the fine detail of the original negative is lost in the half-tone reproduction.

ease in loading and processing, roll film and film pack are excellent, but where great accuracy in the photograph is required, as with the venturi gage, it is generally necessary to resort to plates or cut film, since the more flexible film cannot always be held flat enough. With the venturi gage, the length of the differential often gives a very small value of the magnification and, coupled with the fact that the required readings are in the margins of the field, it is necessary to focus the camera very carefully and to assure that the photographic emulsion be kept flat and accurately positioned in the camera.

In the interests of accuracy and ease of reading, the photographic images should be reasonably large in size, and the cameras should be chosen accordingly. If desired to increase the image size, it is recommended not to draw closer to the subject and thus increase the effects of obliquity, but to use a lens of longer focal length, for it is seen by Equation [3] ($M = f/[U - f] = f/U$, for usual cases) that the magnification is directly proportional to the focal length. The camera must of course be large enough to accommodate the greater image distance and image size.

Of the cameras employed in the pump tests, that used on the venturi gage (f = 16.5 cm) was the least satisfactory, as the recording of the 100-in. differentials was sometimes uncomfortably close to the limiting resolution of the system; a larger size of camera is strongly recommended for differentials of this magnitude. Further, the use of plates is extremely tedious, and a considerable saving in material, time, and patience can be effected by making up a "repeating back" for the camera. This consists of a special holder loaded with a single large piece of plate or cut film, which may be slid along in guides on the back of the camera to give successive partial exposures. Since the image of the venturi gage is essentially long and narrow, this would permit placing several photographs beside each other on the same plate or film, with great economy in handling.

In ordinary linear measurements, increased precision is usually attained by using a more finely divided measuring scale. When the scale has to be photographed, however, there is an optimum spacing of graduation, dependent upon the magnification and type of emulsion; if the divisions are made any finer, it will be found impossible to distinguish the detail satisfactorily. A safe guide, as mentioned previously, is to have the smallest graduation not less than $^1/_{200}$ in. in the photograph. Due to the tendency of image lines to spread slightly, it is desirable to use scales of the "bisection" type (graduation at the imaginary bisector of a line of appreciable thickness, used in most cases) rather than of the "boundary" type (graduation at the boundary line between differently colored spaces of equal width, as in surveyors' leveling rods). The spread of the boundary images tends to give unequal widths to the alternate dark and light spacings of the scale. In order to photograph well, the scale must further have good contrast and preferably a nonglossy surface.

Further items may be tabulated as follows:

1 A prime requisite is a sturdy support for the camera, which may require the construction of special stands, as illustrated in Fig. 1 of the paper.

2 The cameras should always be focused by visual inspection of the image, and the field made large enough to take care of any possible fluctuations in position of the subject.

3 The negative material should be a fast, fine-grain, panchromatic emulsion and be processed under fine-grain conditions. It will generally be found desirable to keep exposure on the low side and to develop to a fairly high contrast to permit easy reading.

4 Since the exposure times must be fairly short, it is necessary to provide good illumination on the meters and gages. This is generally best obtained with overvoltage, high-efficiency lamps (photoflood type) which are commonly available, but as their life is relatively short, some means ("Handbook of Photography," p. 282) is necessary for reducing the input voltage except during the actual time of exposure. If many lamps are in use, care must be taken not to overload the circuits and, in order to reduce the number of lamps to a minimum, they should be housed in efficient reflectors. The illumination units should be compact and easily adjusted in position to concentrate the light where it is most needed and placed, of course, to avoid specular reflection on the subject and direct illumination on the camera lens.

Norman G. McDonald.[11] The photographic method of recording test readings as described and illustrated in this paper has two distinct advantages (1) thereby, a permanent accurate record of the test readings of each instrument is made for future reference, and (2) the personal element is almost entirely eliminated as photographers are substituted for trained engineering observers.

The pumping equipment tested by the author is a part of the $14,000,000 extension to the Toronto Water Works System for which the writer's firm, Gore & Storrie, jointly with H. G. Acres & Company are the consulting engineers on the design, construction, and operation; the specifications for the pumping equipment referred to in this paper were drawn up by them.

The specifications require that each pump with its motor be supplied as a unit with an over-all guaranteed efficiency. The value of 1 per cent efficiency was specified for each unit and the amounts so ascertained were used in evaluating tenders as well as for deductions from the contract payments in case the contractor failed to meet the guarantees. As the value of 1 per cent efficiency on the largest unit amounted to $4450 or $12^1/_2$ per cent of the contract sum, it was necessary that the acceptance tests be carried out as accurately as possible.

Another requirement of the specifications was that the equipment should operate without objectionable noise. As the specific speeds of two of the raw-water pumps were relatively high for the 9-ft suction lift specified, this caused the engineers considerable concern.

The instruments used in these tests were very reliable. The calibrating of meters and instrument transformers, assembled and connected in the same way as used in the tests, undoubtedly improves their accuracy. The use of a water manometer on the venturi-meter tubes and a mercury manometer for reading the pressures reduces the calibrations to a minimum.

The necessity for calibrating venturi tubes under certain conditions has been demonstrated in the paper. The writer has found that, unless preceded by a straight length of pipe of at least 12 pipe diam or other means of insuring straight-line flow, the calculated venturi-tube coefficients are too high and errors up to nearly 10 per cent in the indicated flow have been experienced. The present tendency toward the use of tubes with lower differentials increases the relative error caused by disturbed flows.

Another difficulty in securing accurate flow or pressure measurements is the oscillation or surges in the manometers, as demonstrated by the readings in Table 2 of the paper. Even under the most favorable conditions of flow, wave action is always present, which causes velocities in the piping to the manometers and, due to the inertia of the water, a surge is set up and the manometer readings overrun. The fluctuations in the various instruments do not synchronize and, while trained observers would not record, without comment, the top of a surge on an instrument, this is likely to happen in the photographic method and the reading would be in error.

Damping the connecting pipe lines by partially closing valves is not very satisfactory but the writer has on several tests inserted small orifices in the pipe lines to damp out the surge.

[11] Partner, Gore & Storrie, Consulting Engineers, Toronto, Canada.

Electric meters operate more rapidly than the water manometers but, due to the lightness of the moving parts and the use of air or magnetic damping, there is very little overrun.

The test of the 48,000,000 gal per day unit indicated a pump efficiency of 92.7 per cent as calculated from the over-all efficiency of 88.5 per cent and a motor efficiency of 95.5 per cent. As the venturi tube was carefully calibrated by displacement and the tests made with precision and in duplicate, this high efficiency in the writer's opinion is accurate. The technique described by the author in this paper constitutes a distinct advance in the art of testing centrifugal pumps.

W. S. Pardoe.[12] The venturi-meter air differential gage might advantageously have been of the pot type, described by the writer,[13] thus doing away with the necessity of reading one of the columns and correcting for angularity or parallax.

The author draws attention "to the smoothness and finish of the model," and then uses Professor Moody's step-up formula. Professor Moody has expressed some doubt concerning the applicability of his formula to centrifugal pumps. The use of dimensional analysis presupposes a model and prototype of the same proportional roughness, in which case, the efficiencies would be the same at equal Reynolds' number. If the model and prototype are both smooth and all the losses are frictional, the lost head may be expressed in terms of Chezy's formula and Blasius' expression for f, thus

$$h_f = f\frac{L}{D}\frac{V^2}{2g} = \frac{0.3164}{\left(\frac{VD\rho}{\mu}\right)^{1/4}} \times \frac{L}{D} \times \frac{V^2}{2g}$$

as

$$H \alpha V^2,\ h_f = K\frac{H}{(D\sqrt{H})^{1/4}}$$

and

$$E = \frac{H - h_f}{H} = \frac{H - K\dfrac{H}{(D\sqrt{H})^{1/4}}}{H}$$

or

$$(1 - E) = K\frac{1}{D^{1/4}H^{1/8}}$$

from which

$$\frac{1-E}{1-e} = \left(\frac{d}{D}\right)^{1/4}\left(\frac{h}{H}\right)^{1/8}$$

Professor Moody makes the exponent of $(h/H) = 0.1$. The prototype is usually rougher than the model and some of the losses vary as V^2 instead of $V^{1.75}$, hence the writer suggests that the exponent of $d/D = 0.2$; the revised formula would then be

$$\frac{1-E}{1-e} = \left(\frac{d}{D}\right)^{0.2}\left(\frac{h}{H}\right)^{0.01}$$

This should be used only at the point of highest efficiency and, at other discharges, the peak differential should be multiplied by the ratio of discharges squared.

The writer in deriving the coefficients of venturi meters $d_2/d_1 = 1/2$ used the expression

$$C = \sqrt{\frac{1-(d_2/d_1)^4}{1-(d_2/d_1)^4 + k}} = \sqrt{\frac{1-(d_2/d_1)^4}{1-(d_2/d_1)^4 + \dfrac{0.045}{d_2^{0.23}}}}$$

or $k = \dfrac{0.045}{d_2^{0.23}}$ is the coefficient of loss in the upstream cone, hence his suggestion of 0.2 for the Moody formula.

The author has raised something of a question regarding the venturi-meter coefficients of the A.S.M.E. Fluid Meters Committee, in which the writer is not disinterested, by producing a coefficient of 0.9545 for a 42 × 27-in. venturi meter instead of a normal value of 0.9876. There can be no doubt about the accuracy of his work, backed up as it is by the efficiency of the prototype stepped up from the model. The only reasonable explanation the writer can give for this abnormal coefficient is that the 42,000,000 gal per day pump discharges the water with a considerable whirl, thus lowering the coefficient. The effect of a free vortex[14] is shown in a previous paper by the writer, as well as the length of pipe in which a minor vortex[15] will persist. The writer, therefore, suggests that a traverse be made immediately ahead of the meter to determine if there is a whirl, and if so, straightening vanes should be placed in the pipe to eliminate it.

The writer joins with the author in congratulating the designers and builders of these pumps for their very high efficiencies, rivaling as they do the best of turbine practice, which was thought to be quite impossible only a few years ago.

F. G. Switzer.[16] There are two points in this paper which should be emphasized. The first of these deals with the fact that the manometers under observation during these tests were in constant motion, indicating that there was no damping between the pipe line and the manometer. This is a very desirable practice because damping can easily introduce appreciable error in test results. While this error may not always be very large, there are occasions where it may be quite significant. When the damping device offers different resistance to flow in the in and out directions, definite errors will be produced of unknown magnitude. When the damping device is symmetrical and offers resistance proportional to the square of the velocity, additional errors will be introduced which will be significant only if the pressure variations which cause flow in the manometer connections are large. The only damping which should ever be permitted is what may be called "viscous" damping, in which the resistance to flow is proportional to the first power of the velocity. In order to secure this type of damping, one must of necessity use either a very small-bore tube or else large tubes substantially packed full of needles or something equivalent thereto; both of these arrangements may make it difficult to clear the air out of the connections. The author is to be complimented for having used no damping.

Another comment is in connection with the venturi-meter coefficient. The fact that the coefficient obtained in test is 3.3 per cent below that recommended by the Fluid Meters Committee of the A.S.M.E. is very significant. The writer is of the opinion that much longer approach pipes should often be used than has been customary and that straightening vanes in the approach pipe may be used to reduce this length. As for the meter itself, the construction of the throat has considerable importance. If the piezometer connections are too close to the inlet cone, the meter coefficient will be smaller than anticipated. The shape of the throat passage may also be such as to produce the same results. Some standardization of throat design should accompany the Fluid Meters Committee's report of coefficients, if this has not already been done.

[12] Professor of Hydraulic Engineering, University of Pennsylvania, Philadelphia, Pa.

[13] "Effect of Installation on the Coefficients of Venturi Meters," by W. S. Pardoe, Trans. A.S.M.E., vol. 58, 1936, Fig. 3, p. 678.

[14] Ref. (13), Fig. 32, p. 682.

[15] Ref. (13), Fig. 31, p. 682.

[16] Professor of Mechanics and Hydraulic Engineering, Head of Department, College of Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

W. M. White.[17] The 93 per cent efficiency obtained on one of the pumps is not an impossible efficiency. The filtration pumps in the City of Milwaukee also showed an efficiency of 93 per cent when the quantity of water was measured volumetrically.

A point of interest to the writer is the close adherence of the actual performance of stepup to the theoretical performance of step up by the Moody formula. In the case of the Metropolitan Water District pumps a stepup similar to that at Toronto was not secured. We have as yet been unable to find the reason why the large pumps on the Metropolitan did not give the high efficiency which should have been shown.

One important point not entirely clarified in the paper is the fact that the coefficient of one of the venturi meters was found, by volumetric measurement, to be in error by 3 per cent. That is to say, when the manufacturer's coefficient was used, the quantity was 3 per cent higher than when the coefficient was corrected by means of volumetric measurement. This emphasizes the necessity of careful calibration being made on venturi meters for important centrifugal-pump tests.

Arthur Rynders.[18] The writer's experience shows that the data hardest to obtain accurately in a pump test concern the volume of water pumped.

Some years ago, in testing centrifugal pumping units at the Menomonee Valley Booster Station, the water pumped was measured volumetrically. At this station there are no venturi meters. The pumping units are used to pump water from a six-million-gallon welded steel ground storage tank into the distribution mains. Field measurements of the tank were determined before filling the tank to find as nearly as possible its exact size at a known temperature. During the test, various levels of the water surface were measured by means of a hook gage. The volume computed from these measurements was corrected for the elastic volume change of the tank and the temperature of the tank wall at the time of test. The results of the test were considered to be within satisfactory limits of accuracy.

Recently the pumping units at the Water Purification Plant were tested. The units tested consist of four 50-mgd and one 75-mgd low-level pumping units; and two 20-mgd wash-water pumping units. It was decided to measure the water volumetrically even though venturi meters were available in the low-level discharge line. The tanks were underground reinforced-concrete structures 297 × 372 ft in plan × 22.5 ft deep. Hook gages were again used to determine the water levels. The computed volume was used without any correction. The test results obtained were considered to be very accurate. Calculation made from these tests and motor tests indicated that some of our pumps have an efficiency slightly exceeding 93 per cent.

Author's Closure

In his discussion Mr. Allan suggests that, for large differentials on the venturi meters, one camera for each column would have some advantage, but the author believes that there would always exist some question as to whether the cameras exactly synchronized. The single camera for the two columns gave excellent results and there was much less trouble in setting up the apparatus, changing films, etc.

Mr. Allan mentions the use of motion-picture cameras. Much thought was given the question, particularly in connection with the electrical instruments. However, since the contractors were still working on the building and equipment, the facilities at hand were far from perfect and precluded some methods which might otherwise have been further considered. In work of this kind, it is necessary to obtain the results of tests quickly, which would scarcely have been possible with the motion-picture film. In the actual test, films and plates exposed during one night were developed and available within about 10 hr, so that the result of each test was determined promptly.

The use of the Moody formula in comparing the results on the model and prototype has been referred to several times in the discussion. The author holds no brief for this formula and fully realizes some, if not all, of its defects, although there appears to be no reason why it should not apply to pumps as well as to turbines. The exponents in the head-and-diameter relation, as given by Professor Pardoe may be closer than those given by Professor Moody, and it would seem that experience alone would be the safest guide in this direction. All that the author can say in his defense is that the formula seemed to be well worth trying and it has given quite satisfactory results on these pumps.

In order to compare the results of Professor Pardoe's suggested exponents for the head-and-diameter ratios in the step-up formula, the author recalculated the results on these pumps which are given in Table 7 of this closure. The exponent of H/h is 0.1 as suggested by Professor Moody and 0.125 by Professor Pardoe, while the exponents of d/D are 0.25 and 0.2, respectively.

TABLE 7 COMPARISON OF MOODY FORMULA AND PARDOE MODIFICATION

Pump capacity, million gal per 24 hr	Measured model efficiency e	Calculated efficiency E of prototype: Moody formula	Pardoe modification	Measured efficiency on pump
12	0.900	0.908	0.905	0.902
24	0.896	0.910	0.907	0.915
48	0.896	0.920	0.914	0.930
30	0.907	0.924	0.921	0.918

It is seen that the two calculations give almost the same results but, in this case, the maximum ratio D/d was only 2.63. Since the values of H/h lie between 0.81 and 1.47 it will make slight difference whether the exponent 0.125 or 0.1 is used.

In the cases quoted by Mr. Babb and Dr. White the Moody formula did not prove satisfactory, which shows that it is either defective in construction or that the exponents vary in different cases. The scale of the models mentioned by Mr. Babb was much smaller than those mentioned in the paper, and there may be some little differences due to the method of discharge measurement used.

Comments on the venturi-meter coefficient are interesting, and Mr. Babb's statement that he found differences in the coefficient of 0.3 to 0.4 per cent between calibrations is striking and gives reason for some concern, assuming, of course, that the elapsed time between the calibrations was not great. The variations mentioned by Mr. McDonald are so large that one wonders whether they are due to setting or to manufacture or both. The author has made a careful examination of Professor Pardoe's laboratory and the methods he has used in calibrating meters and believes them to be fully as accurate as claimed. At the same time, the coefficient given by the Fluid Meters Committee, on Professor Pardoe's authority, was undoubtedly in error in the meter discussed in the paper.

When opportunity offers, the author will try to follow Professor Pardoe's suggestion of traversing the pipe close to the meter. In the meantime the author's result rather shakes confidence in the coefficients, undoubtedly accurate for the circumstances under which they were obtained, but uncertain in some field installations. Unfortunately, no suggestion is made as to how one is to know whether or not the Fluid Meters Committee's coefficients should be corrected. The author primarily introduced this matter in

[17] Manager and Chief Engineer, Hydraulic Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

[18] Mechanical Engineer, City Engineer's Office, City of Milwaukee, Wis.

the hope that those interested would follow it up. While one should always strive toward an ideal layout, as suggested by Professor Switzer, he must be governed by practical limitations of space and cost.

Criticism is offered because only one meter was checked, but it must be remembered that the paper describes commercial-acceptance tests, not laboratory experiments. Since the other meters gave no cause for suspicion, they were not checked. The efficiencies of the pumps greatly exceeded the guarantees in each case, and made a calibration of the meters unnecessary. As a matter of fact, there were many difficulties connected with the calibration which rather discouraged the author from attempting all of the meters.

Replying to Professor Daugherty, the author admits he should have stated that the paper referred to field tests, although it hardly seemed necessary. The author's extensive testing practice evidently differs from Professor Daugherty's, for he has scarcely ever found a pump of any size without a venturi meter attached; it is certainly quite common practice in the East. It has been quite usual in the author's work to use a differential gage across the meter, but mercury is nearly always employed in the gage, for practical reasons. However, the cooperation of the contractors and City of Toronto made it possible to use water columns directly; this alone presented some problems but the solution of them brought its own ample reward.

The employment of a mercury manometer for heads as high as 270 ft is also unusual in field tests and the author believes that these refinements produced results of an unusually high order. The author can scarcely understand using uncalibrated electrical instruments, but his calibration on the entire combination of transformers and instruments took care of all errors automatically. There is no statement in the paper that the author accepted the motor efficiencies as given by the manufacturer and he has never done so in any of his work where motor performance was involved. The paper states that the guarantees in the contract were made on the over-all performance of the motor and pump in each case and the separate performance of each part was not required and was not reported on. For this paper, however, the author wished to determine the pump efficiency, and he took the official test results obtained on the motors by an independent expert, not by the manufacturer; there is no reason whatever to question these results.

There seems to be a suggestion that, because the author employed a combination of refinements, each of which had been used in laboratories before, there was no improved technique involved, which is similar to the contention that a new combination of old principles does not constitute a new invention; if such an argument is accepted the Patent Office records will be very brief and very few inventions will be made. Presumably, if Professor Daugherty knew of any other cases where the author's improved technique had been employed in field work, he would have mentioned them. The work represents not only an improved technique, but an accurate and inexpensive method of testing; the cost of assistants and photographs was only about fifteen per cent of what it would have been had direct observers been used.

There are momentary variations in the frequency of the power used in practically all supply lines, as the author knows to his cost in time and patience, but these do not come into the speed calculations of the motors, and the measurement of the slip gives a far more accurate value of the speed of the induction motors than any other practicable method.

The discussion by Mr. Jones of errors possible in the photographic records is very helpful and a good guide as to precautions which must be observed in such work; the numerical results are enlightening. The errors he mentions were either corrected for or eliminated in the tests described.

The discussion by Dr. White draws attention to the very high efficiency of 93 per cent obtained, a result which does not appear to have been exceeded and may at present be regarded as the maximum obtained. Although the water was measured by venturi meter, yet, since the meter had been calibrated volumetrically, the measurement was equivalent to a volumetric one. The author's experience is that volumetric measurements of large discharges in large reservoirs may easily be subject to serious errors, and it was found that unusually great care was necessary in the meter calibration. The time taken in making such measurements would have made the method difficult, if not impossible, in the actual tests. The author's measurements of discharge are as accurate as volumetric measurements and much more easily made.

A Theory of Cavitation Flow in Centrifugal-Pump Impellers

By CALVIN A. GONGWER,[1] DETROIT, MICH.

In this paper the present methods of describing pump cavitation performance are reviewed from the standpoint of the influence of eye design characteristics. From cavitation plots against three eye parameters, cavitation performance is related to eye design and the influence of the angle of attack of the vane leading edges on the flow is studied. Coefficients are derived for the pressure drop due to the vanes and shrouds separately. The parameter *S* or suction specific speed is related to the three eye parameters and interpreted as a design parameter. Cavitation design categories are suggested. The cavitation plots show that the discontinuity in the pump-discharge characteristic curves is due to a "stall" or separation of flow in the eye. A correlation is found between this separation and the angle of attack of the vane leading edges and design information is derived.

Nomenclature

THE following nomenclature is used in the paper:

σ = Thoma-Moody cavitation parameter (1);[2] $\left(=\dfrac{H_{sv}}{H}\right)$

H_{sv} = absolute pump inlet head, less vapor pressure of liquid, and inclusive of velocity head

H = output head or difference between absolute head at pump discharge and absolute head at pump inlet, each inclusive of velocity head

S = suction specific speed (2); $\left(=\dfrac{N\sqrt{Q}}{H_{sv}^{3/4}}\right)$

N = speed, rpm

Q = capacity, cfs

N_s = specific speed; $\left(=\dfrac{N\sqrt{Q}}{H^{3/4}}\right)$

k' = dimensionless constants in total parameter; $H_{sv} \div \dfrac{Q^2}{2g\left(\dfrac{\pi D^2}{4}\right)^2}$; $\left(=\dfrac{2\pi^2}{16}\right)$

C_1 = through-flow index; $\left(=\dfrac{H_{sv}\,D^4 g}{Q^2}\right)$

g = acceleration of gravity

D = impeller eye diameter as determined at narrowest portion of entrance

v_r = radial component of relative velocity

v_t = tangential component of relative velocity

k''' = dimensionless constants in total parameter; $H_{sv} \div \left[\left(\dfrac{\pi N D}{60}\right)^2 \dfrac{1}{2g}\right]$; $\left(=\dfrac{3600 \times 2}{\pi^2}\right)$

C_3 = peripheral-flow index; $\left(=\dfrac{H_{sv} g}{N^2 D^2}\right)$

β = angle of attack of vane inlet edges

α_{i0} = vane inlet setting angle

β_e = complement of angle of flow entry; $(=\text{arc cot}\, k''C_2)$

k'' = constant in total parameter; $\left(\dfrac{4Q}{\pi D^2}\right) \div \left(\dfrac{\pi N D}{60}\right)$; $\left(=\dfrac{240}{\pi^2}\right)$

C_2 = angle of entry index; $\left(=\dfrac{Q}{ND^3}\right)$

K_1 = inlet-shroud coefficient of pressure drop

K_2 = vane-profile coefficient of pressure drop

β_c = angle of attack of vane leading edges at which an inferred separation corresponding to a discontinuity in characteristic curves occurs

Introduction

Nature of Cavitation. The development of cavitation in hydraulic pumps and turbines has been recognized for many years as a major disturbing factor which limits operating ranges, causes destructive pitting, and decreases the efficiency of the machine. The term cavitation is used loosely to describe the formation and violent collapse of vapor or of vapor and gas bubbles formed within the liquid, as a consequence of extreme reductions in the absolute static pressure. It is supposed that this phenomenon influences the head, power, and efficiency of a machine through the decrease in effective fluid density, caused by the presence of vapor bubbles, and through the shock and vibration losses attending the sudden formation and violent collapse of the bubbles as they are passed to regions of higher static pressure. In addition, cavitation pitting and the erosion of passage walls is now widely attributed to the severe mechanical hammering and picking action, resulting from this violent collapse or implosion of the bubbles adjacent to the passage walls.

Current Cavitation Parameters. To facilitate description of the conditions under which cavitation occurred, the Thoma-Moody parameter σ was proposed (1).[2] With respect to pumps, sigma is defined as the ratio of the absolute inlet head (less the vapor-pressure head) to the effective output head

$$\sigma = \frac{H_{sv}}{H} \qquad [1]$$

The convenience of this term obtains from the supposition that, for a particular value of sigma, the aggregate state of cavitation throughout the pump remains essentially constant, provided speed and capacity are varied in the same ratio. In a sense, sigma is the ratio between the excess pressure available for resisting cavitation and the square of the wheel velocities, as measured by the net head across the pump. The net head, through its relation to wheel velocities, may be thought of as representing the ability of the blades to tear holes in the liquid, whereas, the static portion of the inlet head tends to hold that ability in check.

If the inlet head is reduced while the speed and capacity are held constant, experiments demonstrate that the net head de-

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[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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creases little or none at first, but eventually decays rapidly as cavitation becomes general (3). Fig. 1 is typical of the relationship obtained by reducing the inlet head during a laboratory "cavitation run."

The application of sigma is clearly limited to a particular design. For example, if it is conceded that the primary cavitation region is close to the impeller eye, where the lowest static pressures oppose the underpressures due to acceleration of the liquid, as it comes under the influence of the vanes and shrouds,

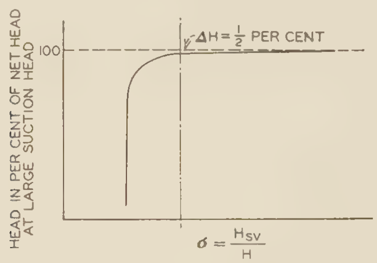

Fig. 1 Conventional Cavitation Plot for Constant N and Q
(H_{sv} = total inlet head above vapor pressure; H = net head across pump at large inlet head.)

then for two impellers of the same eye design and at the same suction head, speed, and capacity, but with different discharge diameters, values of sigma would no longer be the same, although the extent of cavitation could be supposed sensibly identical. In other words, sigma is not independent of the ratio between outside and eye diameters, and is therefore inconvenient for the comparison of eye designs.

G. F. Wislicenus, R. M. Watson, and I. J. Karassik (2), recognizing the need for a more general expression, proposed the "suction specific speed"

$$S = \frac{N\sqrt{Q}}{H_{sv}^{3/4}} \qquad [2]$$

analogous to the familiar specific speed $N_s = \frac{N\sqrt{Q}}{H^{3/4}}$ but with the absolute inlet head above vapor pressure substituted for the usual head H, across the pump; N and Q are the speed and capacity, respectively. The suction specific speed thus measures the extent of cavitation in different impellers of the same eye design, independently of the discharge-to-eye diameter ratio. It may be thought of as equivalent to a sigma corrected to terms of impellers with a common ratio between discharge and eye diameters.

In general, if critical cavitation or breakdown values of the suction specific speed are obtained for impellers of different eye designs, that design with the largest values of critical suction specific speed in a chosen normal specific-speed region would permit operation at the lowest absolute inlet head. In this way, values of the critical suction specific speed become an index of relative merit for any particular operating specific speed.

Need for Definitive Study. In the suction specific speed, however, there is no way of characterizing the flow and eye design conditions which contribute to cavitation. If universal relationships between design and flow conditions can be demonstrated, then, with empirical corrections, it should be possible to use these relationships in interpreting cavitation performance of different specific-speed regions in relation to each other and to design.

A detailed investigation of these possibilities is presented in this paper in conjunction with suggestions for utilizing the relationships developed as guides to improved design.

Three Parameters of Cavitation

Assumptions. A logical approach to a basic analysis of the cavitation conditions is found by consideration of the general case of flow around a submerged streamlined body, such as that shown in Fig. 2. From the physical conditions of flow continuity, fluid incompressibility, and the conservation of energy (Bernoulli's theorem), streamlines may be drawn which give information as to the fluid velocities and, therefore, by the application of Bernoulli's theorem, the pressures at any point in the flow. In short, for the region adjacent to a curved boundary,

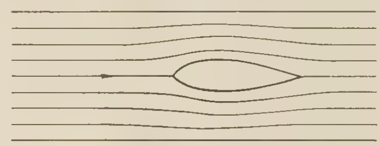

Fig. 2 Flow Around a Streamlined Body

the streamlines are usually close together, and close spacing of the streamlines indicates high velocities and therefore low pressures. By increasing the angle of attack of the body in Fig. 2, the critical region of closely spaced streamlines and underpressure is shifted and increased in magnitude in accordance with the flow laws stated. These general considerations apply to the flow through the impeller and, therefore, definite regions of underpressure which change their position and magnitude with the operating conditions may be inferred. These regions may be considered liable to cavitation.

Implicit in the foregoing general statements there are, in relation to pumps, certain exclusions and assumptions which should be emphasized before proceeding.

1 It is assumed that the region of the impeller eye is characterized by the lowest absolute pressures and, hence, is the region where critical cavitation conditions are likely to obtain. This automatically excludes consideration of disturbances in the casing, at tongue or diffuser vane tips, or at the trailing edges of the impeller vanes. It thus excludes as well the influence of impeller cutting or of changes in the ratio of the discharge to eye diameters and, therefore, implies that the eye flow pattern is not affected by these factors.

2 The effects of the dissolved gases or other impurities in the water are ignored. These factors could alter the effective vapor pressure or the distribution of a cavitation zone in relation both to space and to inlet pressure, but their influence on the experimental results to be cited later was probably small.

3 The effects on the velocity distribution within the impeller passage of any secondary circulation induced by rotation are neglected.

4 It is assumed that the flow pattern through any impeller passage remains the same regardless of the phase angle of the passage as the impeller revolves. This is not strictly correct because uneven pressure distributions around the impeller circumference, under operating conditions other than the design point, accelerate or retard the flow as the impeller rotates. However, because of the inertia of the fluid in the passages and the high frequency of these disturbing pressure changes, the response or velocity variation is probably small.

For convenience in the subsequent analysis, another assumption has been made, and may be listed as follows:

5 The impeller eye diameter is considered to typify, for the purpose of defining peripheral velocities, the diameter of the imaginary surface of revolution described by the leading edges of the impeller vanes. As the diameter of a circular area, it is also assumed to define the effective radial entrance area of the vane passages. These assumptions are considered justified

because this circular area is very close to the vane inlet edges at the eye periphery. Since here the peripheral and relative velocities are greatest, this can be considered as the critical cavitation region.

The Through-Flow Index, C_1. As a corollary to the discussion of the general case of flow around streamlined objects, the underpressures may be seen to be proportional to the entering velocity head. A measure of the pressure available for resisting cavitation is the entering pressure head less the vapor pressure. The ratio of these two quantities, the entering pressure head less the vapor pressure to the entering velocity head, should therefore provide a measure, for geometrically similar flow patterns, of the proximity of the flow to the cavitation regime.

The entering velocity is nearly the capacity divided by the eye area. The ratio of the heads is therefore, $H_p \div \left[\left(\frac{4Q}{\pi D^2}\right)^2 \times \frac{1}{2g}\right]$. However, since the total inlet head rather than the pressure head is more generally considered in pump practice, we will consider only the ratio of the total inlet head to the entering velocity head. The two ratios are connected by the equation

$$H_p \div \left[\left(\frac{4Q}{\pi D^2}\right)^2 \frac{1}{2g}\right] + 1 = H_{sv} \div \left[\left(\frac{4Q}{\pi D^2}\right)^2 \frac{1}{2g}\right] = k'C_1 \quad [3]$$

where k' is introduced to describe the dimensionless constants hereinafter disregarded, C_1 is the dimensionless parameter, Q is the flow in units of volume and time, g is the acceleration of gravity, and D is the diameter of the impeller eye. The parameter

$$C_1 = \frac{H_{sv}D^4g}{Q^2} \quad [4]$$

will be called the through-flow index of cavitation.

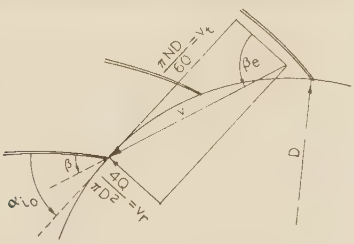

FIG. 3 VELOCITY DIAGRAM OF IMPELLER-VANE LEADING EDGES

If the flow pattern remains geometrically similar regardless of actual velocities, a high value of the through-flow index means that the pressure is high relative to the velocity head, and that the possibility of cavitation is remote. A low value indicates the reverse.

The Peripheral-Flow Index. By applying this reasoning for the general flow case to the case of the impeller eye, it can be seen that another velocity may be used as a measure of the tendency of the liquid to cavitate. This velocity is the absolute velocity of the vanes at the eye periphery and, in accordance with the fifth assumption, is $\frac{\pi ND}{60}$. The derived measure of the proximity of the flow to the cavitation regime is, therefore

$$H_{sv} \div \left[\left(\frac{\pi ND}{60}\right)^2 \frac{1}{2g}\right] = k''' C_3 \quad [5]$$

in which k''' again represents the dimensionless constants to be ignored, and

$$C_3 = \frac{H_{sv}g}{N^2D^2} \quad [6]$$

is the dimensionless parameter which it is convenient to retain. To signalize its relation to the tangential eye velocity, C_3 is called the peripheral-flow index. The peripheral-flow index is a companion to the through-flow index and describes the proximity of the flow regime to a condition of cavitation.

Angle-of-Entry Index. The two cavitation parameters now derived are measures of the proximity of the flow regime to a condition of cavitation only for geometrically similar flows, for only in this case are the velocities proportional in magnitude and equal in direction. In other words, the validity of the indexes as cavitation criteria holds only for one operating point on the pump characteristic curves. In Fig. 3 may be seen a simplified velocity vector diagram at the impeller eye. In accordance with the fifth assumption, the relative through-flow or radial velocity component is $4Q \div \pi D^2$ and the relative tangential velocity component is $\pi ND \div 60$. If the speed is held constant and the capacity Q is varied, the resultant relative velocity v, which is the vector sum of the through-flow and peripheral velocities, changes in direction according to the relationship

$$\cot \beta_e = \frac{4Q}{\pi D^2} \div \frac{\pi ND}{60} = k''C_2 \quad [7]$$

where k'' is again the dimensionless constant, and is $240 \div \pi^2$, and

$$C_2 = \frac{Q}{ND^3} \quad [8]$$

As may be seen from Fig. 3, for a given vane setting α_{i0}, β, or the angle of attack of the flow on the vanes is defined as

$$\beta = \alpha_{i0} - (\pi/2 - \beta_e) = \alpha_{i0} - \tan^{-1} k''C_2 \quad [9]$$

It is β, the angle of attack, which is the criterion of flow similarity and, since, for a given vane setting, α_{i0}, β is measured by $\tan^{-1} k''C_2$, C_2 is taken as a more useful parameter of flow similarity. Since C_2 is the measure of inlet flow angle, it will be called the angle-of-entry index.

EMPIRICAL ANALYSIS OF CAVITATION PERFORMANCE

Parameters Related to Performance. A procedure for associating the three indexes with the intensity or degree of cavitation condition in terms of the extent of its influence on pump performance is indicated by the testing practice of the pump laboratory at the California Institute of Technology. With closed-circuit facilities, the pump capacity and speed are held constant while the inlet pressure is progressively decreased. Fig. 1 illustrates typically the effect of such inlet-pressure reductions upon the net output head of the pump. For the particular speed,

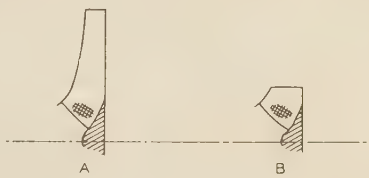

FIG. 4 RELATIVE IMPORTANCE OF CAVITATION ON NET HEAD BETWEEN IMPELLERS OF LARGE AND SMALL OUTSIDE DIAMETERS

capacity, and design tested, the drop in the curve from right to left, therefore evaluates the extent of cavitation development in terms of pump performance. Two degrees of decay in pump performance frequently referred to in the laboratory are the "complete breakdown," where the curve drops vertically, and a partial breakdown where the head loss, from some arbitrary starting point, is 0.5 per cent.

In order to divorce the "one half of one per cent" point from the variations of volute design and impeller outside-to-eye

diameter ratio, the author has found it advisable to refer the head loss to the head of twice the eye peripheral velocity $(2v_t)^2 \div 2g$, where v_t is the impeller eye velocity, $\pi ND \div 60$. This choice of reference head is arbitrary and is approximately the head corresponding to a ratio of outside-to-eye diameter of 2. Inspection of Fig. 4 gives some clue as to the reason for this practice. Two impellers A and B, with identical eyes but different outer diameters are shown. If the same cavitation degree occurs in both, presumably under the same values of the cavitation indexes, as represented symbolically by the shaded areas, the resulting head drops referred as percentages to the eye velocity heads, should be the same. It has been found that this method eliminates the difference in the total heads from consideration. Henceforth, the "half per cent points" will be those eye operating points at which the head loss across the pump in feet ΔH, referred to the head of twice the eye peripheral velocity $(2v_t)^2 \div 2g$, is 0.5 per cent

$$\Delta h = {}^1/_2 = \frac{\Delta H}{(2v_t)^2 \div 2g} \times 100 \qquad [10]$$

Presentation of Experimental Data. The utilization of the three indexes in describing the complete cavitation characteristics of a particular eye design is shown diagrammatically in Figs. 5 and 6. The two charts, one of C_1 against C_2 and the other of C_3 against C_2, tell essentially the same story. In plotting the charts, the indexes have been evaluated for the complete break-

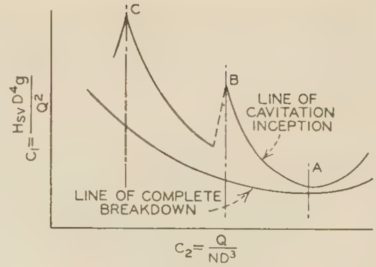

FIG. 5 COMPLETE EYE CAVITATION CHARACTERISTICS; C_1 VERSUS C_2

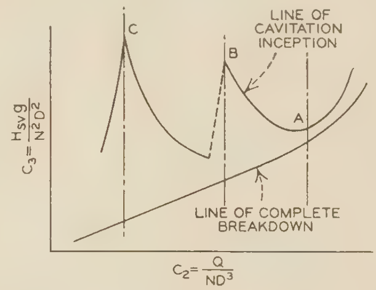

FIG. 6 COMPLETE EYE CAVITATION CHARACTERISTICS; C_3 VERSUS C

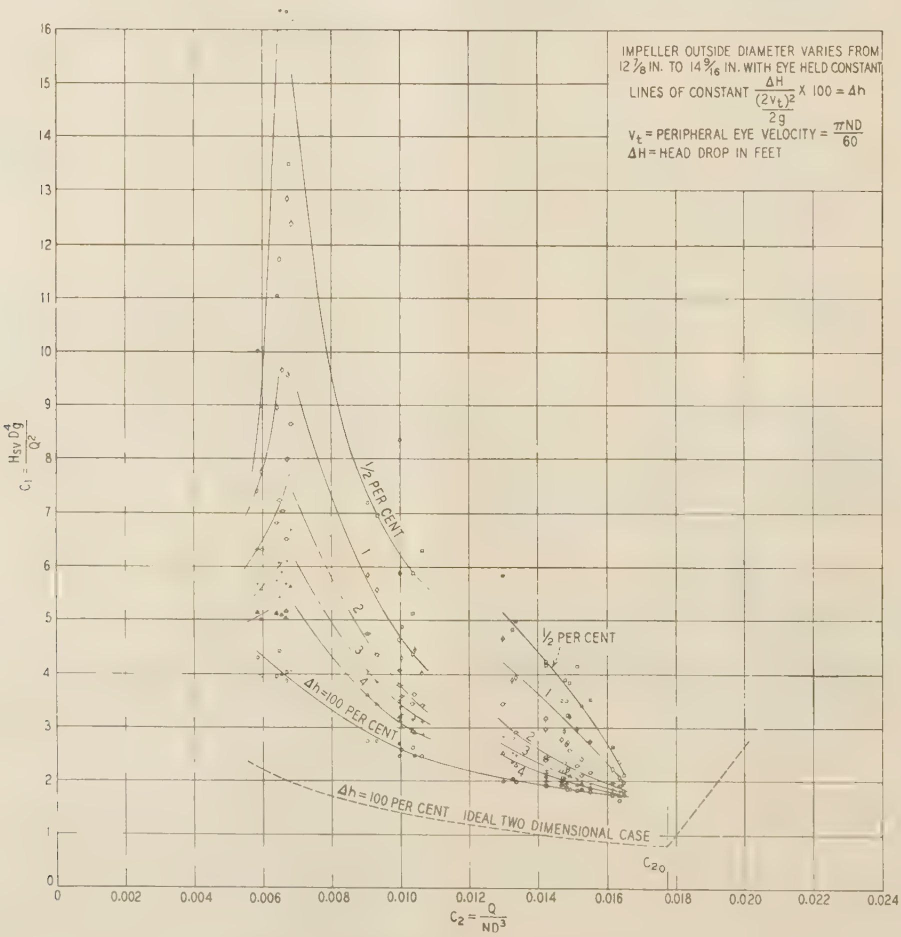

FIG. 7 DIMENSIONLESS CAVITATION PLOT

off and the 0.5 per cent point, here labeled the line of cavitation inception, for a number of cavitation runs taken, as described, at different values of C_2 over the pump range of capacities. From measurements of the vane setting, point A has been found to correspond to zero angle of attack of the vane leading edges in accordance with Equation [9]. The close spacing of the two lines, of cavitation inception and breakoff, at point A corresponds to a very sharp breakoff in the head as the suction head is lowered. The discontinuities at B and C have been found to occur for the pumps tested wherever the data have been taken over a sufficient range of capacities and will be discussed later.

In Figs. 7, 8, and 9, actual cavitation data are plotted in this form. The first two plots are for several impellers with the same eye design and widely differing outside diameters. Since, however, the same foundry pattern was not used for all the impellers and slight differences in vane thickness, setting, and spacing unavoidably occurred, there is a slight scatter in the points of the breakoff curve. The scatter is seen to become progressively greater as the degree of cavitation decreases and the reason for this is apparent from Fig. 1 in the low slope of the head curve at high suction heads. However, careful inspection of these points has shown that the differences in results among the impellers of different outside diameters is of small order with respect to the scatter for any one impeller and, therefore, the plots have been considered to represent eye performance only. The importance of this fact is emphasized, particularly in the light of the interpretation which is put on point B, Figs. 5 and 6, in the discussions which follow.

From the nature of the three cavitation indexes, it is possible to represent in the general cavitation plots the complete characteristics of the particular eye design, and regardless of eye diameter (scale), speed, or capacity, C_2 is the same measure of flow similarity and C_1 and C_3 are the same measures of the cavitation regime. The general utility of this type of chart is therefore great, particularly for the designer.

Evaluation of Eye Coefficients From Experimental Data. In Equation [9] the expression for the angle of attack β was derived

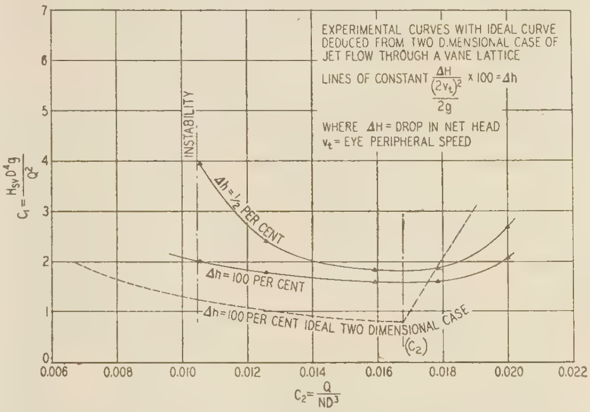

FIG. 9 IMPELLER CAVITATION CHARACTERISTICS

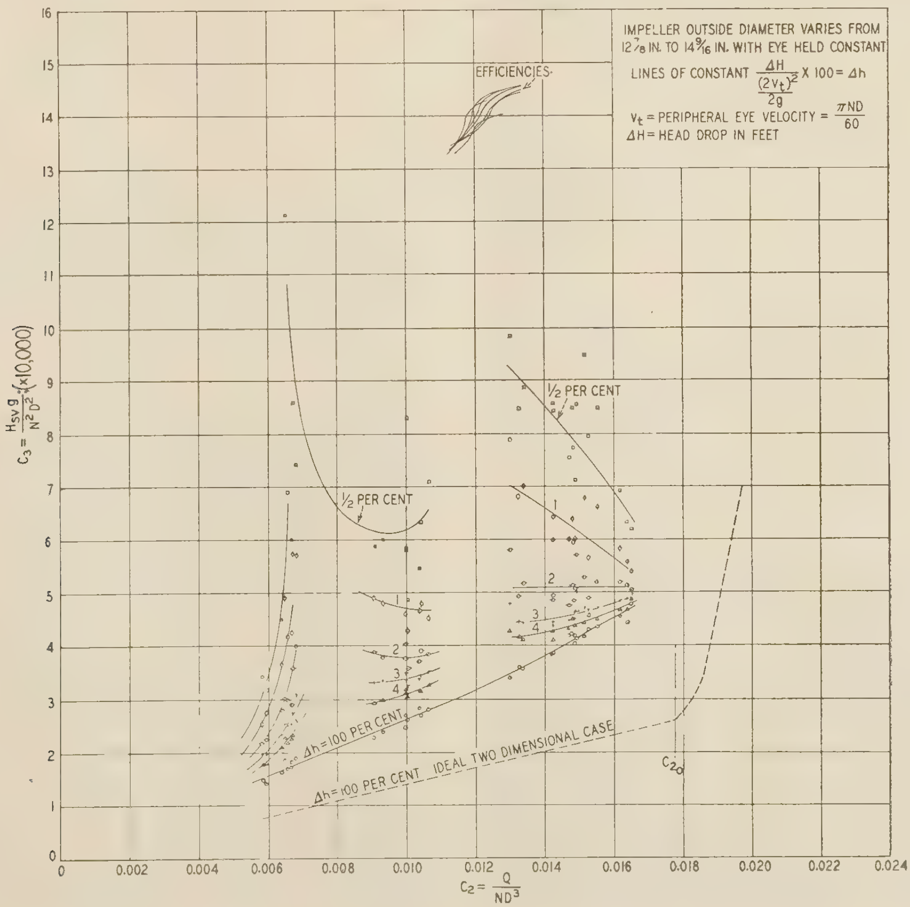

FIG. 8 DIMENSIONLESS CAVITATION PLOT

by inspection of Fig. 3, in terms of the vane setting α_{i0}, and C_2, the angle-of-entry index. By setting β equal to zero in this equation, it is seen that the zero angle-of-attack point can be represented by the expression

$$\tan^{-1} k'' C_{20} = \alpha_{i0} \qquad [11]$$

where C_{20} is the value of C_2 corresponding to zero angle of attack of the vane leading edges. Therefore, if α_{i0} is varied from low to high values, by varying the vane setting at the leading edges, C_{20} will also vary from low to high values and point A in Figs. 5 and 6 will correspondingly move from left to right. However, only by taking separate eyes with differing vane settings can this variation in C_{20} be effected.

In connection with operation at C_{20}, the pressure drops in the eye can be expressed by coefficients pertaining to the particular part of the passage in question. For instance, the maximum pressure reduction attending the flow around the vane can be expressed as a coefficient times the head of the relative velocity, v in Fig. 3. To obtain the total maximum underpressure in the eye, this vane pressure drop can be added to the pressure drop effected by the shrouds in guiding the flow from the axial to the radial direction. The latter underpressure, in conventional eyes, would logically occur at the suction-side shroud about at the junction with the vane leading edges and, hence, the two underpressures are superimposed as a first approximation.

According to the mechanics of cavitation assumed in the development of the parameters C_1 and C_3, the maximum underpressure in the eye equals the pressure of the entering flow at the critical cavitation or breakdown point. This can be expressed by an equation due to Spannhake and Wislicenus

$$H_p = m \frac{\left(\dfrac{4Q}{\pi D^2}\right)^2}{2g} + n \frac{v^2}{2g} \qquad [12]$$

where m and n are the shroud and vane coefficients, respectively, applied to their corresponding velocity heads; or by adding the through-flow velocity head to both sides and substituting for v its value from Fig. 3

$$H_{sv} = K_1 \frac{\left(\dfrac{4Q}{\pi D^2}\right)^2}{2g} + \frac{K_2}{2g}\left[\left(\frac{4Q}{\pi D^2}\right)^2 + \left(\frac{\pi N D}{60}\right)^2\right] \qquad [13]$$

where H_{sv} is the total absolute suction head above the vapor pressure, $K_1 = m + 1$ and is the new shroud coefficient, $K_2 = n$ and is the vane coefficient. By dividing by the through-flow velocity head, we have

$$k'C_1 = K_1 + K_2\left[1 + \frac{1}{(k''C_{20})^2}\right] \qquad [14]$$

where k' and k'' are defined in Equations [3] and [7]. The subscript zero is applied to C_2 to signify the zero angle of attack.

By inspection of Equation [14], it can be seen that the cavitation performance at the design point C_{20}, can be calculated from the two parameters K_1 and K_2. It thus is advisable to attempt to evaluate these coefficients by whatever method possible and, barring a rigorous evaluation, to use an approximate one. To use a rigorous method, it would be necessary to make several impellers with different vane settings C_{20}, but with identical vane and shroud profiles and to adjust the values of K_1 and K_2 until the cavitation performance of all these eyes could be accurately described at the design point by Equation [14]. Since this method has not been practicable, the less rigorous procedure was followed in striking a mean line through the breakoff points of all the impellers tested. These varied widely in vane setting and were assumed to vary not too widely in profiles of vanes and shrouds so that the mean constants K_1 and K_2 could be evaluated. This mean curve is shown as curve B in Fig. 10 with each point representing a particular eye with vane setting C_{20}. The scatter is encouragingly small in view of the variety of impellers represented. The constants were determined by trial and error until, on substitution in Equation [14], curve B obtained. The values are 1.4 and 0.085 for K_1 and K_2, respectively. With these values Equation [14] may be written

$$k'C_1 = 1.4 + 0.085\left[1 + \frac{1}{(k''C_{20})^2}\right] \qquad [15]$$

This equation is of practical importance and can be used to predict cavitation performance at C_{20} with the precision allowed by the scatter on curve B in Fig. 10. The values of K_1 and K_2 are considered to be well within the range expected from the nature of the flow around these profiles.

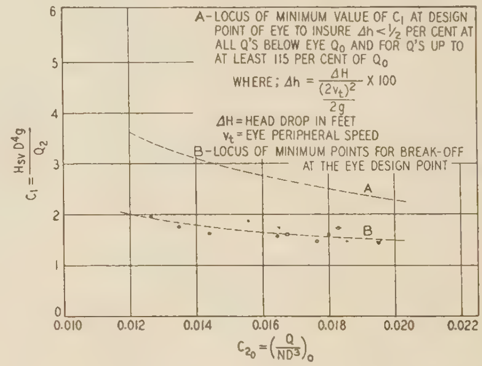

Fig. 10 Curves of Locus of C_1 and of Locus of Minimum Points for Breakoff

A Theoretical Limit Function. Although an empirical analysis of the cavitation performance at C_{20} has been effected without much difficulty, the determination of the nature of the breakdown phenomenon at finite positive or negative angles of attack is inherently more difficult because of the interference between the vanes. It is desired to derive some sort of theoretical curve to approximate the breakoff curves in Figs. 7, 8, and 9. Whereas, the breakdown case for all the impellers at zero angle of attack has been considered, the breakdown curve for a particular impeller at all angles of attack will now be investigated. Here, the interference effects will involve consideration of a theoretical lattice of vanes in a flow, which for simplicity sake will be considered two-dimensional.

Before proceeding, it is necessary to postulate a picture of breakdown cavitation at finite angle of attack. Since here the head across the pump drops practically to zero, it is reasonable to assume that the vanes exert little or no lift on the fluid. This is possible only if the fluid breaks completely away from the suction side of the vanes and is accelerated in a free jet which proceeds along the pressure face to the discharge into the volute. This picture has been assumed by Betz and Petersohn (4) and considered analogous to the case of a two-dimensional lattice of straight, thin vanes of finite width in a rectilinear flow. This ideal picture is shown in Fig. 11, where the region on the suction side of the vanes contains only vapor at the corresponding low absolute pressure.

This two-dimensional case has been mathematically analyzed (4) by means of conformal mapping and the fundamental equation results

$$\cos(\beta - \theta) = \frac{(R + 1/R)\cos\beta + (1/R - R)\sin\beta\tan\beta e}{2} \quad [16]$$

where β is the angle of attack, θ is the angle made by the deflected free jets as they emerge from the lattice, with the oncoming undisturbed flow direction, $R = v_1/v_2$ where v_1 and v_2 are the undisturbed relative approach and free-jet velocities, respectively, and β_e is the angle of the oncoming flow with the line joining the vane leading edges. These angles can be seen in Fig. 11.

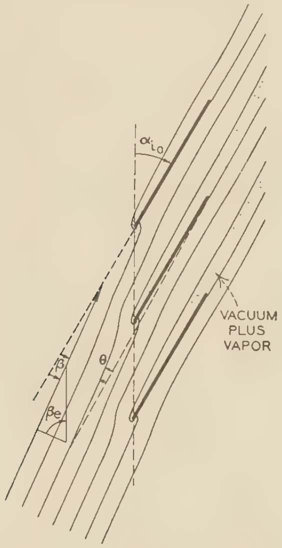

Fig. 11 Breakdown Cavitation in a Two-Dimensional Lattice

In order to use Equation [16], a simplifying assumption must first be made in accordance with the large chord-to-gap ratio occurring in impellers. It is necessary to assume that for this case the free jets of Fig. 11 leave the vanes in a direction parallel to the vanes. From the picture it is clear that θ is then equal to β and the left-hand member of Equation [16] is unity. With this simplification, values of R as a function of β may be determined for a particular vane setting α_{i0} since Equation [16] reduces to

$$R = \frac{\cos\beta_e \pm \sin\beta}{\cos(\beta + \beta_e)} \quad [17]$$

Once R is known, the performance can be calculated by assuming that the total energy head of the oncoming relative flow is converted to velocity at zero pressure head in the free jets. The resulting equations are

$$H_{1p} + \left(\frac{4Q}{\pi D^2}\right)^2 \frac{1}{2g} + \left(\frac{\pi ND}{60}\right)^2 \frac{1}{2g} = v_2^2 \frac{1}{2g} \quad [18]$$

therefore, by substitution

$$H_{sv} - \left(\frac{4Q}{\pi D^2}\right)^2 \frac{1}{2g} = \frac{1}{2g}\left[\left(\frac{4Q}{\pi D^2}\right)^2 + \left(\frac{\pi ND}{60}\right)^2\right](1/R^2 - 1)$$

and by transposing and dividing by the through-flow velocity head

$$k'C_1 = 1 + \left[1 + \frac{1}{(k''C_2)^2}\right](1/R^2 - 1) \quad [19]$$

Thus, given R as a function of β, and of C_2, from Equation [17], Equation [19] determines $k'C_1$. The resulting ideal curves are plotted in Figs. 7, 8, and 9 as dashed lines. These curves fall far below the actual curves since K_1 and K_2, the vane and profile coefficients, have been automatically taken as unity and zero, respectively, in this analysis, which is for thin vanes with no curvature in the third dimension such as that imposed by the shrouds.

The practical value of the ideal breakoff curves lies in the fact that their shape is similar to the empirical break-off curves. The sharp upturn at overcapacities, negative values of β, now seems to be due to the severe interference effect of the stagger as β passes from positive to negative. It is felt that this similarity in shape justifies the picture of breakdown cavitation assumed in Fig. 11.

It has been attempted to raise the ideal curve to the empirical curve by adding the increments due to the difference between the actual values of K_1 and K_2 and the assumed values, unity and zero. While good agreement has been found occasionally, more often it has been poor, and it is felt that the assumption of two-dimensional flow so limits the analysis that the actual flow picture is no more than qualitatively described.

Compatibility With Aeronautical Experience. In the field of aeronautics, it has been found that the interference between the wing and the fuselage at the suction side of the wing on low-wing planes has necessitated the use of fillets to prevent separation of the flow in the region of the re-entrant angle. This separation consists of eddies toward the wing trailing edge. It is believed that a similar case is encountered in the impeller eye in the re-entrant angle, where the suction side of the blade joins the suction shroud at the eye periphery. Here the underpressures due to the shroud and vane profiles can be thought of as augmenting each other. For this reason cavitation would be expected to occur first in this interference region. What is thought to be the effect of smooth fair fillets can be seen in the flatness of the cavitation inception line of Fig. 9, which shows test results from an impeller with such fillets. Fig. 7 represents an impeller without these fillets and it can be seen that the cavitation inception line is very steep near the design point, which indicates there are large underpressures at small angles of attack.

That this region of the blade span next to the suction shroud at the eye periphery is critical for cavitation is further evidenced by the observed fact that α_{i0} must be measured as the angle made by the vane trace in the suction shroud with the circle of revolution, if the measured value of C_{20} from Equation [9] is to correspond with the experimental value of C_{20} for both cavitation and complete breakdown.

Aeronautical practice can be borrowed further in considering the question of sharp or blunt vane leading edges. The frequent use by aeronautical engineers of the "teardrop" or blunt-nosed profile for fuselage and wing sections for optimum drag characteristics would at first seem to suggest the use of blunt-nosed impeller vanes, particularly since the underpressures on these profiles are less sensitive to angle of attack. However, as previously suggested in connection with flow around curved boundaries, the higher curvatures induce large local underpressures which, from Bernoulli's theorem, must correspond to high local velocities. This proves the undoing of the blunt-nosed profile for aeronautical purposes if these local velocities approach that of sound through the fluid, for the induced standing sound waves bear witness to the inability of the fluid to adjust itself with sufficient rapidity to conform with the required flow pattern. There is an analogy between the sonic velocity for the airplane section and the cavitation velocity, or the velocity corresponding to the total absolute suction head, for the pump-vane section. It has been observed experimentally that sharp vanes give better cavitation performance both for inception and complete break-

off, since the cavitation velocity of the total absolute suction head is reached at lower suction heads for the sharp vane sections.

Eye Design From Empirical Data

Values of S Related to New Parameters. The parameter S, or suction specific speed, as presented by Wislicenus, Watson, and Karassik (2) is related to C_1, C_2, and C_3 in the following manner:

$$S = \frac{N\sqrt{Q}}{H_{sv}^{3/4}} = \frac{1}{\left(\frac{C_1}{g}\right)^{3/4} C_2} = \frac{1}{\left(\frac{C_2}{g}\right)^{1/2}\left(\frac{C_1}{g}\right)^{1/4}} = \frac{1}{\left(\frac{C_3}{g}\right)^{3/4}} C_2^{1/2} \quad \ldots . [20]$$

Referring to the locus of eye design points in Fig. 10, we see that those impellers to the left correspond to high design S, while

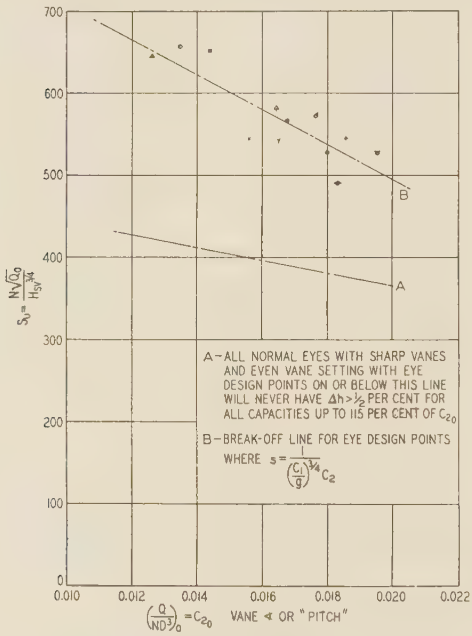

Fig. 12 Replot of Fig. 10

those to the right correspond to low values of S. This means that low S corresponds to large α_{i0} or steep vane settings, and high S to flat vane settings, where S is applied to the eye design point C_{20}, as N_s, or discharge specific speed, is applied to the pump peak efficiency point.

Using the second of Equations [20], the curves in Fig. 12 can be derived from those of Fig. 10. Curve B of Figs. 10 and 12 is a definite line as may be seen from the relatively small scatter of points. Curve A is the author's recommendation for a conservative design limit and, when applied to the eye design point, gives the conditions under which Δh will be less than $^1/_2$ per cent, up to 115 per cent of the design-point capacity. The suction heads specified by curve A are conservative by 20 per cent for good eyes and radical by 10 per cent for poor ones.

Design Example. The curves of Fig. 12 are useful for design and performance prediction purposes, as will be illustrated by the following example:

Given design operating conditions:

$$Q_N = 7 \text{ cfs}; \quad N = 2000 \text{ rpm}; \quad H_{sv} = 32 \text{ ft abs}$$

therefore

$$S_N = \frac{2000\sqrt{7}}{32^{3/4}} = 394$$

From curve A for $S = 394$ in Fig. 12, it is apparent that $C_2 = 0.0163$ and, therefore, the vane setting can be determined from Equation [11]

$$\frac{240}{\pi^2} 0.0163 = \tan \alpha_{i0}; \quad \alpha_{i0} = 21.5 \text{ deg}$$

The eye diameter can now be determined

$$C_{20} = \frac{Q_N}{ND^3} = 0.0163$$

by definition

$$D^3 = \frac{7}{2000 \times 0.0163} = 0.2145; \quad D = 0.598 \text{ ft}$$

Suggested Design Categories. Eye conditions can be divided into three classes, depending upon the value of S; (*a*) $360 < S < 430$; (*b*) $S < 360$; (*c*) $S > 430$. For class (*a*) eyes, it is recommended that the design be from curve A in Fig. 12, as in the example. For class (*b*) much leeway is allowed in the vane setting and eye diameter because of the adequate suction head. For class (*c*), inspection of Figs. 7, 8, and 9 shows that cavitation-free operation can be secured up to line B, if Q does not vary far from Q_N. The problem of design, therefore, becomes one of compromise among a number of factors to secure the best over-all pump performance. In general, it can be said that for severe conditions the eye vanes should be given a flat setting with consequent large eye. However, there is a lower limit to vane setting as will be discussed hereafter. The best cavitation performance at Q_N is secured by making $C_{20} = Q_N/ND^3$ or $\beta = 0$, but this does not give much overcapacity to the pump as the interference between vanes causes the cavitation performance to become very

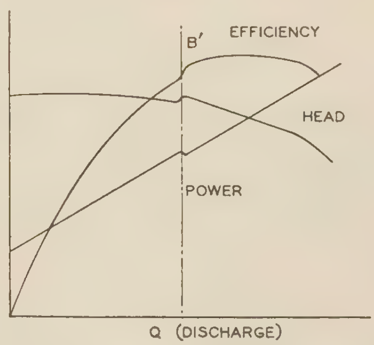

Fig. 13 Typical Constant-Speed-Pump Characteristic Curves

poor for negative values of β. Therefore, it is usually advisable to have β about +1 to 3 deg at Q_N.

Pump Performance in Relation to Separation

Correlation With Empirical Discontinuity. The complete cavitation characteristic plots provide an excellent means of studying the flow conditions in the impeller entrance passages. Certain studies have been made from these plots and will now be presented.

The discontinuities in the cavitation inception lines shown in Figs. 5, 6, 7, and 8 can be traced to the pump characteristic curves and, in particular, point B of Figs. 5 and 6 corresponds to point B' of Fig. 13, which shows typical characteristic curves for a pump with volute-type casing. Experimental confirmation of this correlation is presented in Fig. 8. This figure, as previously explained, represents cavitation tests on a series of identical impellers at different outside diameters. Short sections

TABLE 1 CRITICAL VALUES FOR IMPELLERS

	Group I											Group II				
Impeller	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
β_c, deg	7.1	7.4	7.7	7.9	7.5	7.0	7.5	7.0	7.5	8.0	7.3	2.5	4.1	2.1	2.8	

of the efficiency curves containing the discontinuities have been plotted to the same abscissas. Although, because of an insufficient number of points, this jump in efficiency is not usually noticed in manufacturer's pump tests, it represents a difference of from 1 to 4 per cent.

Since it has not usually been possible for the designer to predict the location of this so-called instability, it seems desirable to pursue further this correspondence linking the instability with the impeller eye. The location of B' has been found to be independent of volute type and impeller cutting and, since it is characterized by a torque change as well as head change, it is more conclusively traced to the impeller and to the impeller eye.

The Mechanics of Separation. From the discussion of the excessive underpressures, resulting from the interference between the suction side of the vane and the suction shroud, particularly at high values of β, it is reasonable to expect separation of the flow at this point. The term separation, as distinct from cavitation, is applied to the sudden departure of the streamlines from a path along the walls and their replacement by dead water or back eddies in the intervening space (5). It is generally held that separation is a consequence of the presence of insufficient energy in the boundary layer to maintain flow against the abrupt pressure increases which follow underpressures. It should be kept in mind that the foregoing remarks refer to noncavitation flow as will the remainder of the argument.

It therefore follows that the occurrence of this separation is accompanied by a large reduction in the underpressures because of the reduction in curvature of the streamlines, and therefore the cavitation sensitivity is reduced. This is verified by the experimental data in Figs. 7 and 8 where a drop in the inception lines with decreasing C_2 occurs at the instability point.

Since the phenomenon of separation is related to the pressure distribution in the flow, which is in turn most conveniently described in terms of angle of attack, in the case of airfoils, it should be instructive to evaluate the angles of attack for which the separation occurs. This has been done with the aid of Equation [9] and Table 1 lists the critical values for all the impellers considered.

There is a striking constancy in angle for all impellers from 1 through 11 with a sudden radical departure for the impellers thereafter. These latter impellers came under the heading of "sick pumps" because of the corresponding narrow capacity range in which the efficiency is maintained at the higher values.

It appears that the impellers of the first group are affected by an almost identical phenomenon, upon which much light can be shed by again borrowing from aeronautical experience. Hovgard (6) describes a premature stall (separation) at the wing-root leading edge of some of the more recent airplanes. This stall results from interference with the fuselage, and means of eliminating it were devised. It was found that the fore-and-aft location of the wing was the principal factor. It is believed that the instability of the impeller is due to a similar effect, by which is meant a stall at the leading edges near the suction shroud. If this is the case, the methods of alleviating this condition, following the aeronautical case, should consist of flattening the setting slightly in the immediate vicinity of the shroud and raking the leading edge forward out of the eye.

It is instructive to conjecture as to the nature of the seemingly different separation occurring in the latter group of impellers. It can be said of these impellers that the exit vane settings were exceptionally flat. This flat setting is believed to have resulted in a tendency to separate at the suction shroud independent of the vane leading edges. This separation would result in a sheet of dead water covering the entire suction shroud. Pitot traverses at the impeller discharge, at capacities on either side of the instability, have suggested that this might be the case. It can be shown that the lift of the vanes tends to rotate this dead water from the suction shroud to the suction side of the vanes and thus suppress the separation. It is therefore suggested that, in the absence of sufficient vane steepness, the low vane lift allows an entirely different separation, to control which may be responsible for the performance of the second group of impellers.

With the value of β_c given in group I of Table 1, the author has been able to predict the location of the instability point from measurements of the eye vane setting. The setting is measured at the eye periphery in the separation region.

Generalizations

Cavitation Study. Several general conclusions may be drawn from the cavitation study:

1 Cavitation performance is a function of eye vane leading-edge and shroud profiles and, consequently, the eye cavitation performance can be calculated from coefficients characteristic of these profiles together with the vane setting angle and eye diameter.

2 The complete cavitation characteristics of an impeller eye design can be described in terms of the dimensionless parameters C_1, the through-flow index C_2, the angle-of-entry index, and C_3, the peripheral index.

3 For optimum cavitation performance, the vane leading edges should be sharp and well filleted where they join the shrouds.

4 The term S, the suction specific speed, can be made a design parameter by plotting empirical values against C_2, as has been done in Fig. 12.

5 The term S stipulates the eye vane angular setting if suction conditions are severe, i.e., poor conditions corresponding to high S require flat vane settings and large eye, while good conditions corresponding to small S permit leeway in vane setting.

Separation Study. The general conclusions which may be drawn from the separation study include:

1 An assumed eye separation is the cause of the discontinuities occurring in the discharge characteristics and this separation is of two types (*a*) a local leading-edge stall adjacent to the suction shroud, and (*b*) a full suction-shroud separation independent of the vanes and which is a result of insufficient vane lift (low exit vane setting).

2 Given the eye angle setting, the eye diameter should be adjusted, cavitation conditions permitting, to locate the instability point entirely out of the operating region.

Acknowledgment

The author is indebted to the staff of the California Institute of Technology and to the United States Bureau of Reclamation for the use of the experimental material upon which much of the argument of this paper is based, and for their generous technical advice and assistance.

The author is particularly indebted to Professors Th. von Kármán, R. L. Daugherty, and R. T. Knapp and to Mr. J. W. Daily of the Institute and to Mr. D. P. Barnes of the U. S. Bureau of Reclamation.

Special thanks are due Dr. George F. Wislicenus of the Worthington Pump and Machinery Corporation and Mr. A. Hollander, chief engineer of the Byron Jackson Company for valuable tech-

nical assistance and permission to publish test data from their respective machines.

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Discussion

R. W. Angus.[3] The material presented in this paper is of timely interest and the author has taken a method of analysis of the factors affecting cavitation in pump impellers that is most constructive. The centrifugal pump has come into very common use but, unfortunately, numbers of them cause trouble through noise and vibration, at times accompanied by a rapid deterioration of the pump. This noise does not necessarily indicate an inefficient pump, for some of those with highest efficiency are noisy and decidedly objectionable from the standpoint of the operator.

After the pump is installed, the operator is greatly concerned with the destructive effect of the noise and, if he is not running the pump continuously, he may not be able to tell whether erosion is serious or not until after the pump has been paid for. Noise is sometimes eliminated by admitting a very small volume of air to the suction pipe, and often there is no appreciable loss in efficiency thereby. Designers are in need of such studies as outlined in this paper, so they may be sure their pumps will be satisfactory. The writer believes, however, that the design of the eye is not the sole cause of the trouble, but that the form of casing and the design of the inlet passages are also important factors.

The writer is not clear as to the connection between the curves in Figs. 6, 7, 8, and 10 of the paper and would appreciate more detail. For example, Fig. 7 gives no definite indication of the point B shown in Fig. 5 and, while there is a suggestion of a point like B in Fig. 8, it is by no means as distinctly marked as in Fig. 6. What was the experimental background for suggesting the presence of the point B, and why is it suggested in Figs. 5 and 6 that the curves turn up to the right of A when, in Figs. 7 and 8, most of them are pointing downward?

In Fig. 12 the author does not clearly state his reason for drawing the curve B in the position shown. Assuming the curve to be properly placed, the investigation shows how it may be employed in design to good advantage.

The horizontal axis $\dfrac{Q}{ND^3}$ of the curves is proportional to the square of the specific speed for a given pump, since ND represents the linear speed of a point on the impeller and is related to the head produced by the latter; Fig. 12 thus relates two specific speeds both connected with the conditions at entry to the impeller.

The types of curves, shown in Fig. 13, are not uncommon and the discontinuities have been noticed by many engineers. Too often they have been ascribed to irregularities in the tests themselves, and a smooth curve drawn through the points in such a way as to mask these sudden changes of curvature. The writer believes that the pump characteristics are very sensitive to the actual setting of the pump, and has noticed that pumps not infrequently appear to be quite satisfactory when tested in the shops, but show considerable trouble when placed in service, even under apparently good suction conditions. Wherever possible, the water should enter the pump under some pressure. In future investigations, the writer would suggest trying the effect of elbows of different radii and set at various angles somewhere in the suction line, not necessarily close to the pump.

G. F. Wislicenus.[4] In order to understand the paper in its relation to other publications in the same field, it seems advisable to establish clearly the relation between the coefficients introduced by this paper and those already in use,[5] as follows

Gongwer $$C_1 = \frac{H_{sv}D^4g}{Q^2};\quad C_3 = \frac{H_{sv}g}{N^2D^2};\quad C_2 = \frac{Q}{ND^3}$$

Bibliography (2) of paper $$S_q = \sqrt{\frac{g}{C_1}};\quad S_n = \sqrt{\frac{g}{C_3}};\quad \frac{S_q}{S_n} = C_2$$

Sedille $$\mu_0 = \frac{K'''C_3}{2};\quad \delta_0 = \pi \times K'' \times C_2$$

It can be shown by the methods of dimensional analysis that the factor C_1, C_3, and C_2 or their equivalent expressions, together with the suction specific speed S, are the only independent dimensionless coefficients which can be formed out of the variables Q, H_{sv}, N, D. No publication therefore can be expected to introduce fundamentally new coefficients derived from these basic variables. It is rather the better understanding and improved application of these fundamental coefficients to which this paper presents a valuable contribution.

The paper seeks to establish design characteristics rather than performance characteristics. For the designer, the complete coefficients $K'\,C_1$, $K'''\,C_3$, and $K''\,C_2$ will be more useful than the abbreviated coefficients C_1, C_3, and C_2, in particular if the complete coefficients are expressed as simple head and velocity ratios. Putting $\dfrac{\pi ND}{60} = V_t$ (Fig. 3 of paper) and $\dfrac{4Q}{\pi D^2} = V_E$ (equivalent to V_r in Fig. 3) we obtain

$$K'C_1 = \frac{2gH_{sv}}{V_E^2};\quad K'''C_3 = \frac{2gH_{sv}}{V_t^2};\quad K''C_2 = \frac{V_E}{V_t}$$

This form has the added advantage of extending the significance of the given equations to pumps with their shafts running through the eye of the impeller, while the same coefficients and relations in the form given in the paper apply to single-suction overhung pumps only.

Equation [13] of the paper permits the derivation of a maximum obtainable value of the suction specific speeds S for given values of the coefficients K_1 and K_2. An increase in eye diameter D for a fixed capacity Q and speed of rotation N will reduce the absolute fluid velocity in the eye and, thereby, the first term of

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[4] Worthington Pump & Machinery Corporation, Harrison, N. J. Mem. A.S.M.E.

[5] Discussion Bibliography (2) of paper, Trans. A.S.M.E., vol. 62, Feb., 1940, p. 160.

Equation [13], while it increases the relative velocity at which the water meets the vanes and, thereby, the second term of Equation [13]. Obviously, there must be some optimum value of D giving the smallest sum of these two terms and, thereby, the lowest value for H_{sv}. This is obtained by putting the derivative of H_{sv} with respect to D in Equation [13] equal to zero. Using $K_1 = 1.4$ and $K_2 = 0.085$ one obtains for the cavitation breakdown of single-suction overhung pumps $S_{\max} = 740$, (Q in cfs).

Condensate and process pumps have been operated successfully at S values considerably in excess of the maximum value derived in this way. Such operation, however, is not free from considerable cavitation, although mostly without harmful effects. This condition probably prevents the application of Equations [13] and [14] of the paper to these types of pumps.

The application of the results by Betz and Peterson to the cavitation flow in centrifugal pumps is of great interest and the qualitative agreement between the theoretical and experimental results seems to be much better than could be expected, taking into account not only the three-dimensional character of the real flow but also the fact that the simplifying assumption $\theta = \beta$ is questionable because of the rotational character of relative flow in radial-flow runners. The sharp break in the theoretical curve seems to be in error judging from Equations [17] and [19] and from physical reasoning. The quantitive difference, on the other hand, seems to be most easily explainable from the thickness of the vanes in real pumps.

The analysis of the relation between instabilities in the pump head and efficiency curves and the cavitation performance is believed to constitute the most original contribution made by this paper. Its importance is not diminished by the fact that the practical application of this relation so far may be restricted to impellers of the type investigated, so that generalizations of these results do not seem to be within immediate reach. It is rather the general significance of these investigations with respect to the mechanism of the flow inside pump runners which may be far-reaching and which, for this reason, will be discussed.

The writer previously presented a simple theoretical approximation for the beginning of separation on impeller vanes.[6] This approximation was based on the average deflection of the fluid by the vane as a whole (average lift coefficient) and the average pressure rise along the vanes, assuming that the flow conditions in the boundary layers on impeller vanes are essentially the same as those along single aerofoils as tested in the wind tunnel. The results of the present paper on the other hand seem to indicate that not the average flow conditions but those near the inlet edges of the vanes have a predominating influence upon separation phenomena in the runner. The most probable reason for this discrepancy is the fact that the boundary-layer flow in radial-flow pump impellers may differ materially from that observed on single aerofoils in the wind tunnel. Figs. 14 and 15 of this discussion indicate why such a difference in boundary-flow conditions is likely to exist.

Fig. 14 shows separation as observed on a single aerofoil in the wind tunnel. The arrows in the zone of separation represent the average direction of rotation in the boundary layer after the point of separation. Fig. 15 shows the corresponding relative flow conditions in a radial-flow pump impeller, assuming that the direction of rotation in the separation zone remains the same as before. This direction of rotation, however, is opposite to the general tendency of relative rotation in radial-flow runners, indicating that the rotation of the runner tends to suppress separation. This somewhat intuitive reasoning can be confirmed by a simple calculation of the influence of the Coriolis forces on the boundary flow in radial-flow runners.

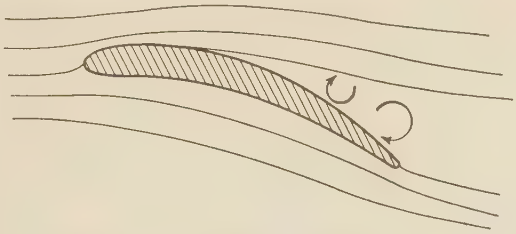

Fig. 14 Separation as Observed on Single Aerofoil in Wind Tunnel

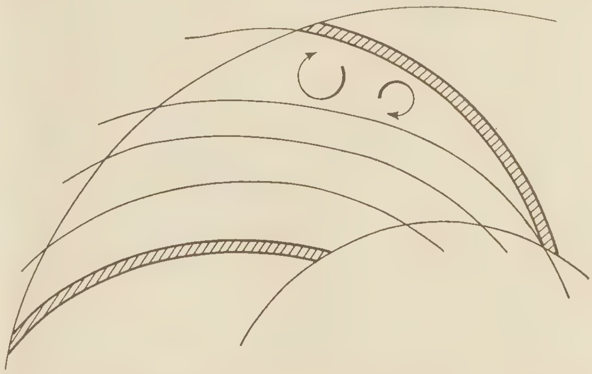

Fig. 15 Corresponding Relative Flow Conditions in Radial-Flow Pump Impeller

The turbulent shearing stress in a two-dimensional boundary layer (such as along an aerofoil in the wind tunnel) usually is expressed in the form

$$\tau_x = \rho \times \overline{V_x' \times V_y'} \quad [21]$$

where V_x' and V_y' are the turbulent-velocity fluctuation in the direction of the fixed boundary x and at right angles to the fixed boundary y; ρ is the mass of the fluid per unit volume.

The corresponding expression for the turbulent boundary layer along the vanes of a radial-flow runner was found by the writer to be

$$\tau_x = \rho \left(\overline{V_x' \times V_y'} + 2\,\omega\,\overline{V_y'\,l}\right) \quad [22]$$

where ω is the angular velocity of the runner, and l some form of a "mixing length" of the turbulent boundary layer. The new term in Equation [22], as compared with Equation [21], represents the influence of the Coriolis forces (or relative rotation) on the turbulent shearing stress. Since these stresses tend to suppress separation, it is apparent that Equation [22] has the same physical meaning as the result previously derived by inspection of Figs. 14 and 15.

If it is assumed that the mixing length l is proportional to the distance between the vanes, the previously mentioned discrepancy between the results of this paper and earlier results of the writer is largely explained, qualitatively confirming the results of the present paper. At the same time it becomes clear that investigations of this type can be expected to throw considerable light on the general problems of flow conditions in hydraulic runners. It is, therefore, believed that this paper deserves the most careful and serious consideration.

[6] "Separation in Pumps and Turbines," by G. F. Wislicenus. Presented at Summer Meeting, University of California and Standford University, June 19–21, 1934, of the Aeronautic and Hydraulic Divisions of The American Society of Mechanical Engineers.

Author's Closure

In answer to the questions of Professor Angus concerning the difference between the experimental curves of Figs. 7, 8, and 9

and the type curves presented in Figs. 5 and 6, it was perhaps not made sufficiently clear that the experimental data for any one impeller were incomplete. The curves for all the impellers could be pieced together qualitatively to show that the type curves of Figs. 5 and 6 obtain over the complete capacity or angle of entry range. Although the uprise is not shown in Figs. 7 and 8 since points are not available at C_{20} (point A in Figs. 5 and 6) and beyond, in every impeller for which data were available in this region the curves turned up as in Figs. 5 and 6. Space considerations prevented the presentation of the many curves for which the discontinuity was more noticeable than in Fig. 7. However, a replot of these data in Fig. 8 shows the discontinuity and its relationship to the efficiency jumps. This correlation between the efficiency jumps and cavitation discontinuity was observed in every instance.

In Fig. 12 the curve B has been drawn through the experimental points shown. Curve A is a rough indication of the maximum height of the curves on the plots of the type of Fig. 8 in terms of the location of C_{20}. This latter curve is offered as a qualitative aid to the determination of cavitation inception limits.

The author wishes to emphasize that the horizontal and vertical variables in Fig. 12 are not the same quantities. The horizontal variable is best thought of as corresponding to the angular vane setting at the eye, and is proportional to tan α_{io} in Fig. 3. The vertical variable is the suction specific speed at C_{20}, or point A in Figs. 5 and 6, which is obtained by eliminating the diameter D from any two of the coefficients C_1, C_2, and C_3. The data are also plotted in Fig. 10 with the ordinate altered from Equations [20].

Regarding the effect of casing design on cavitation performance, the tests for Figs. 7 and 8 were made with impellers cut from $14^9/_{16}$ in. to $12^7/_8$ in. and used in the same casing. Consequently, the head and capacity of the pump varied over wide limits but the eye cavitation performance remained the same within the limits of scatter in the figures. The scatter may perhaps be laid to this impeller cutting which was in effect a change in casing had the impeller remained uncut.

In regard to the discussion of Dr. Wislicenus, the derivation of the maximum value of S to be expected from Equation [13] is very enlightening. However, in comparing this derived value with practice for condensate pumps, it must be remembered that the conditions are stipulated in Equation [13] that operation be at C_{20}, the zero-angle-of-attack point. Inspection of Fig. 8 shows that, for a given speed and eye, H_{sv} for breakoff approaches zero at shutoff. Consequently, S reaches large values toward shutoff since H_{sv} enters as the three-fourth power in the denominator and Q as only the one-half power in the numerator. Therefore, a pump can be operated at practically infinite S if the capacity is low enough. Consequently, this apparent disagreement between theory and practice is not real.

The author feels that the sharp break in the theoretical curves of Figs. 7, 8, and 9 is compatible with physical reasoning for the two-dimensional-flow picture assumed, but the curvature of the actual flow in two additional directions and deviations from a rectangular entrance velocity profile would prevent resolution of this break in the experimental results.

The author agrees with Dr. Wislicenus in his interesting discussion of the effect of Coriolis forces in suppressing separation from the impeller vanes. Similar conclusions can be reached by considering the increased radial head build-up in a dead water region, for backward sloping vanes, which would result in transverse pressure gradients tending to force the live water into the separation region.

Turbulence and Energy Dissipation

By A. A. KALINSKE,[1] IOWA CITY, IOWA

This paper incorporates a study of the origin and dissipation of turbulence energy which makes possible a better understanding of the mechanics of energy losses that are introduced by various flow-disturbing devices such as expansions, bends, valves, etc. The important parameters which characterize turbulence are the root-mean-square values of the fluctuating velocity components, the length factor proportional to the size of the small eddies responsible for the dissipation of energy, and the length factor proportional to the average size of the eddies. The effect of variation of these parameters on the energy losses occurring in turbulent flow are discussed, and also the change in these parameters in the decaying turbulence beyond turbulence-producing devices is indicated. Data are presented showing the variation in the kinetic energy of mean flow and the turbulence energy in a 15-deg conical divergence. Visual studies of the start of turbulence at rounded entrances to smooth conduits seem to indicate that there is a regular vortex formation at the boundary, and the dispersion of these vortexes into the main fluid stream gradually establishes normal turbulent flow.

THE flow of all real fluids involves energy loss due to the frictional resistance of the fluid. The terms "energy loss" or "energy dissipation" are used in this paper to describe the eventual changing into heat of the energy producing flow. Energy dissipation in a continuous fluid for either laminar or turbulent flow is due to the action of the viscosity of the fluid, and such terms as "shock loss" or "impact loss" do not describe correctly how energy losses occur in continuous fluids.

Hydrodynamics indicates quite definitely how energy is dissipated in a fluid in viscous flow. The rate at which energy is dissipated in viscous flow can be predicted from a knowledge of the flow pattern and the characteristics of the fluid. However, this is not the case for turbulent flow. It is ordinarily possible by the use of semiempirical formulas to calculate the total over-all pressure drop and thus the total rate of energy loss, however, we know very little about the mechanics of how this energy loss takes place in the turbulent fluid itself. The purpose of this paper is to analyze more thoroughly the mechanics of energy dissipation in a turbulent fluid, and to relate such analyses to various practical hydraulics problems. Liberal use will be made of the fundamental ideas regarding the mechanism of turbulence which have been put forth by L. Prandtl, G. I. Taylor, Th. von Kármán, and their co-workers. The principles of the statistical theory of turbulence will be extended to the practical problems of energy dissipation in liquid flow.

Fundamental Conceptions

In order that the meaning of various terms and expressions used in this paper may be clearer they will be defined at this stage.

When a fluid flows in a straight conduit at distances far enough beyond bends or transition sections, so-called normal conditions become established. In such a case, the amount and nature of turbulence present is unvarying and the shape of the mean-velocity distribution remains constant. For this condition of flow, the rate of creation of turbulence is in equilibrium with its rate of dissipation. The kinetic energy of mean flow per pound of fluid flowing is ordinarily designated as $\overline{U}_m^2/2g$, where $\overline{U}_m$ is the mean velocity in the cross section. The total kinetic energy of the mean flow is then equal to $Q\gamma\overline{U}_m^2/2g$ where Q is the discharge and γ is the specific weight. Because of the variation in mean velocity, all particles of fluid passing any section do not have the same kinetic energy, therefore the foregoing expression can only be exactly correct for uniform velocity distribution. For a circular cross section, the correct expression for the total kinetic energy of mean flow is

$$E_m = \pi\rho \int_0^r \overline{U}^3 y dy \qquad [1]$$

where
ρ = unit density of fluid
y = distance from center
r = radius of cross section
$\overline{U}$ = mean velocity with respect to time at point y

In true turbulent flow, the velocity at a point varies irregularly with time in direction and magnitude. However, in vortex motion, as for example in the von Kármán vortex street produced beyond a body immersed in a flowing fluid, the variation in the velocity vector with time is quite regular. Whenever the term "turbulence" is used it will refer to a condition of flow where there is no regularity in the variation of the direction and magnitude of the velocity vector, except in the probability sense. It is convenient to designate the velocity vector at any instant by the components U, V, and W along the x, y, and z axes, respectively. The x axis is in the direction of mean flow, the y axis is normal to the mean flow, and the z axis is the other normal axis.

The kinetic energy per pound of fluid at any instant is actually equal to $U^2/2g$. Separate the fluctuating component U into two parts, such that $U = \overline{U} + u$, where u is the fluctuating part. Obviously, $\bar{u} = 0$; (the bar indicating an arithmetic mean). Thus the mean kinetic energy is in reality equal to $(\overline{U} + u)^2/2g$ or to $(\overline{U^2} + \overline{u^2})/2g$ and the part $\overline{u^2}/2g$ is called the kinetic energy of turbulence. Therefore, the total kinetic energy due to the turbulence per pound of fluid will be

$$e_t = \frac{\overline{u^2} + \overline{v^2} + \overline{w^2}}{2g} \qquad [2]$$

The total kinetic energy of turbulence for the fluid flowing in a circular conduit will then be

$$E_t = \pi\rho \int_0^r \overline{U}(\overline{u^2} + \overline{v^2} + \overline{w^2}) y dy \qquad [3]$$

Another important fundamental idea is the concept of shear in a moving fluid. For viscous flow the shear per unit area at any point is equal to $\mu\, d\overline{U}/dy$, or the product of the coefficient of viscosity and the velocity gradient. The mean shear in a turbulent fluid is equal to

$$\tau = \mu \frac{d\overline{U}}{dy} + \rho\epsilon \frac{d\overline{U}}{dy} \qquad [4]$$

The term ϵ is the so-called diffusion coefficient having the dimen-

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sions of a velocity times a length, and a good physical interpretation of ϵ is given by Bakhmeteff (1).[2] In fully developed turbulent flow, except near a smooth boundary, the coefficient $\rho\epsilon$ is many times greater than μ, thus the first term in Equation [4] becomes quite small compared to the second term.

The rate at which potential or pressure energy is used in producing flow is equal to $\tau(d\bar{U}/dy)$ in foot-pounds per unit volume of fluid. For viscous flow this becomes $\mu(d\bar{U}/dy)^2$, and represents the rate at which the energy producing flow is dissipated into heat at any point in a fluid. In fully developed turbulent flow, since the mean shear stress is not influenced by viscosity, the quantity $\tau(d\bar{U}/dy)$ represents the rate at which the pressure-energy producing flow is transformed into turbulence energy. The turbulence energy is then dissipated into heat by the action of viscosity on the small eddies. However, energy of turbulence is not necessarily dissipated at the point where it is created; it can and is diffused by the mixing action of the turbulence. For instance in a uniform conduit, the term $\tau(d\bar{U}/dy)$ is greatest near the boundaries and is zero at the center. The energy of turbulence produced at the boundaries is in part diffused to the center of the conduit and dissipated there. Of course it should be remembered that, in nonchanging uniform flow, the total rate at which energy of turbulence is created is equal to the rate of dissipation of this energy by the action of viscosity.

Energy Considerations for Isotropic Turbulence

Because of the complexity of the turbulence mechanism, any theoretical considerations must start with the simplest conditions. One of the types of turbulence that has received extensive consideration by physicists interested in wind-tunnel turbulence phenomena is that referred to as isotropic turbulence. This type of turbulence is such that (a) the mean values of squares and products of the fluctuating velocity components, such as $\overline{v^2}$ and $\overline{uv}$, and their derivatives, such as $\overline{\left(\frac{\partial v}{\partial y}\right)^2}$ and $\overline{\left(\frac{\partial v}{\partial x}\frac{\partial u}{\partial y}\right)}$ are independent of the location of the point observed, and (b) the same mean values are obtained if the axes of reference are rotated or reflected. Fluids having isotropic turbulence can have no mean shear or mean-pressure gradient, that is, $d\bar{U}/dy = 0$, $d\bar{U}/dx = 0$, etc.

In practical hydraulics, isotropic turbulence is hardly ever attained; however, it is sometimes approached. The turbulence in the center of closed conduits is nearly isotropic; also, the turbulence formed downstream from various turbulence-producing devices such as screens, grids, expansions, and flow-control apparatus tends to approach isotropy at times. As G. I. Taylor remarks, "there is a strong tendency to isotropy in turbulent motion." A study of the energy dissipation characteristics of isotropic turbulence may throw considerable light on the more complicated turbulence obtained in many practical hydraulics problems.

The intensity of turbulence is usually designated by the ratios $\sqrt{\overline{u^2}}/\bar{U}$, $\sqrt{\overline{v^2}}/\bar{U}$, and $\sqrt{\overline{w^2}}/\bar{U}$, and for isotropic turbulence these are all equal. Investigations by Dryden (2) and others reveal that these ratios at any point beyond any particular turbulence-producing device tend to be independent of the mean velocity. This relationship also appears to hold for nonisotropic turbulence at high Reynolds numbers. The decrease in the intensity of the turbulence beyond screens and grids in wind tunnels, or the decay of the turbulence, has been studied quite extensively both theoretically and experimentally by Taylor (3), von Kármán (4), and Dryden (2).

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Lamb (5) shows that the general equation for the rate of dissipation of energy per unit volume in a viscous fluid is

$$K = \mu\left[2\overline{\left(\frac{\partial U}{\partial x}\right)^2} + 2\overline{\left(\frac{\partial V}{\partial y}\right)^2} + 2\overline{\left(\frac{\partial W}{\partial z}\right)^2} + \overline{\left(\frac{\partial V}{\partial x} + \frac{\partial U}{\partial y}\right)^2} + \overline{\left(\frac{\partial W}{\partial y} + \frac{\partial V}{\partial z}\right)^2} + \overline{\left(\frac{\partial U}{\partial z} + \frac{\partial W}{\partial x}\right)^2}\right] \quad [5]$$

In isotropic turbulence U, V, and W can be replaced by u, v, and w. Taylor simplifies Equation [5] by demonstrating the interrelationship existing between the various mean squares and mean products of the velocity gradients, and obtains

$$K = 7.5\mu\overline{\left(\frac{\partial u}{\partial y}\right)^2} \quad [6]$$

Instead of $\overline{\left(\frac{\partial u}{\partial y}\right)^2}$ the term $\overline{\left(\frac{\partial v}{\partial x}\right)^2}$ can be used, since both are equal in isotropic turbulence. In order that a better conception might be had of a term such as $\overline{\left(\frac{\partial u}{\partial y}\right)^2}$ Taylor showed that

$$\overline{\left(\frac{\partial u}{\partial y}\right)^2} = \frac{2\overline{u^2}}{\lambda^2} = \frac{2\overline{v^2}}{\lambda^2} \quad [7]$$

where λ is a length proportional to the small eddies present since they are primarily responsible for the dissipation of the turbulence energy. Possible methods of experimentally determining $\overline{v^2}$ and λ were discussed by the author in a previous paper (6).

The mechanism by which the small eddies are produced from the larger ones is a fundamental problem of turbulence about which little is known. It is these small eddies, referred to sometimes as the microturbulence, which are largely responsible for the high rate of energy dissipation associated with turbulent flow.

The scale of the turbulence, L, which is important in regard to the diffusive action of the turbulence, is proportional to the average size of the eddies. The product $\sqrt{\overline{v^2}}L$ is proportional to the transverse diffusion coefficient ϵ, as used in Equation [4]. As the turbulence created by some obstruction in a fluid stream is dissipated downstream, a change in the length factors λ and L takes place which is a characteristic phenomenon of decaying turbulence. Qualitative visual observations of the turbulence in water streams beyond grids, throttled valves, sudden expansions, etc., seem to indicate that the average size of the eddies tends to increase as the turbulence is dissipated.

The internal stresses in turbulent flow are proportional to the product of the density and the mean square of the fluctuating velocities such as $\rho\overline{u^2}$. Any such force will then dissipate energy at a rate proportional to the product of the force and the associated velocity, thus, $\rho(u')^3$, where $u' = \sqrt{\overline{u^2}}$. The total area on which these forces act will be proportional to the square of the scale of the eddy system, or to L^2, and the rate of dissipation per unit volume will then be proportional to $\frac{\rho(u')^3}{L}$. For isotropic turbulence, this quantity should then be proportional to the rate of energy dissipation given by Equation [6]. The following proportionality can then be written

$$\frac{\lambda}{L} \propto \sqrt{\frac{\nu}{Lu'}} \quad [8]$$

$$\frac{\mu}{\rho} = \nu = \text{kinematic viscosity}$$

The terms under the radical in Equation [8] have been referred

to as a Reynolds number of turbulence. In decaying turbulence, certain early experiments in wind tunnels seemed to indicate that the length factor L tended to remain constant for a considerable distance downstream from the turbulence-producing grid. However, later experiments and analyses indicate that L increases as the dissipation progresses, and also it appears that λ increases faster than L. The value of $\sqrt{\overline{u^2}}$ decreases practically hyperbolically with the distance from the turbulence-producing device. Since the term $\sqrt{\overline{u^2}}$ decreases and λ increases,

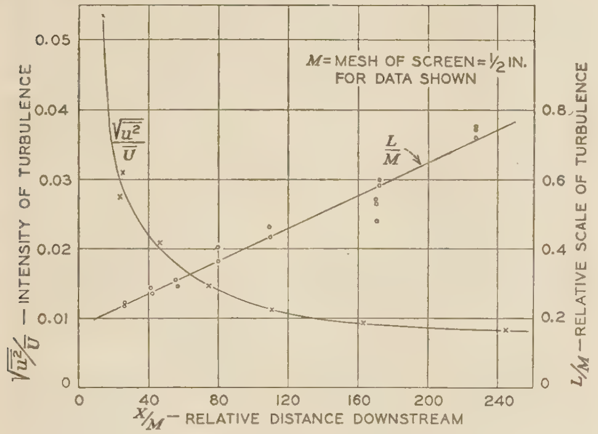

FIG. 1 DATA ON DECREASE IN INTENSITY AND INCREASE IN SCALE OF WIND-TUNNEL TURBULENCE BEYOND A SCREEN

the rate of dissipation of the turbulence energy continually decreases. In Fig. 1 is shown a plotting of some data obtained by Dryden (2) and his co-workers in regard to the decay of turbulence and the increase of the scale factor L beyond a $^1/_2$-in-mesh screen in a wind tunnel. The term L was defined as $\int_0^\infty R_y dy$, where R_y is the correlation coefficient, $\frac{\overline{uu_y}}{\overline{u^2}}$. The terms u and u_y are the simultaneous velocities at two points a distance y apart.

SOME APPLICATIONS OF FOREGOING CONCEPTIONS

The use of some of the ideas relating to energy of turbulence, particularly its creation and dissipation, gives an insight into many practical fluid-flow problems. Probably the simplest form of turbulent flow is that of uniform, steady flow in a closed conduit; yet there is much that we do not understand regarding the mechanism of the dissipation of energy in such a simple case of fluid flow. If the pressure drop is known, the average boundary and internal shear can be accurately determined, particularly for a simple conduit such as one having a circular cross section, or a wide rectangular one where the end effects are negligible. Knowing the shear and the distribution of mean velocity, the rate at which pressure energy is converted into turbulence energy per unit volume is equal to $\tau d\overline{U}/dy$ (assuming fully developed turbulent flow). For a two-dimensional conduit of width $2b$, the shear varies linearly from the value τ_0 at the wall to zero at the center, and we have for this rate of energy conversion

$$\frac{\tau_0}{b} y \frac{d\overline{U}}{dy} \qquad [9]$$

Taylor (3) presents some interesting calculations on the variation of this quantity and the rate of turbulence energy dissipation as calculated from Equation [6] for the dissipation of isotropic turbulence. The calculated values are shown in Fig. 2. The general variation of the rate of pressure-energy conversion, as given by Equation [9], and the rate of dissipation of the energy of turbulence seems to be consistent. Of course, the dissipation formula cannot be accurate near the wall, since isotropy is not approximated in this region. Very near the wall, especially in the boundary layer, much of the pressure energy is converted into heat directly without passing through the intermediate stage of turbulent energy.

The total rate of energy dissipation from the center of this

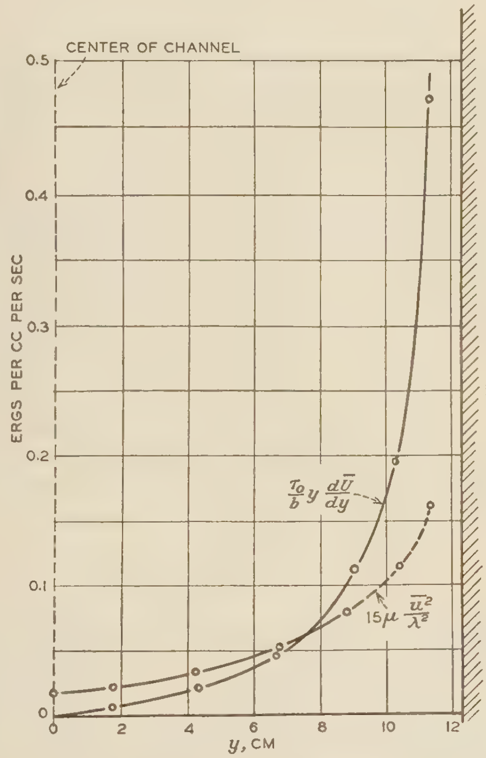

FIG. 2 VARIATION OF RATE OF PRESSURE-ENERGY CONVERSION AND TURBULENCE-ENERGY DISSIPATION PER UNIT VOLUME ACROSS A CONDUIT

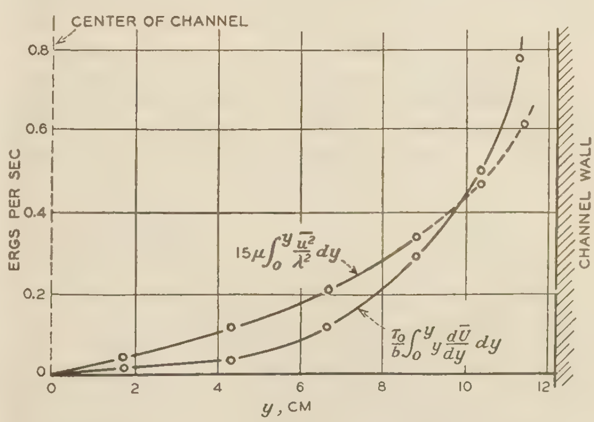

FIG. 3 VARIATION IN TOTAL RATES OF PRESSURE-ENERGY CONVERSION AND TURBULENCE-ENERGY DISSIPATION, AS CALCULATED BETWEEN THE CENTER AND Y DISTANCE FROM CENTER IN A CLOSED CONDUIT

rectangular conduit to some distance y from the center is equal to (Equations [6] and [7])

$$15\mu \int_0^y \frac{\overline{u^2}}{\lambda^2}\, dy \qquad [10]$$

The rate of pressure-energy conversion in the same region is equal to

$$\frac{\tau_0}{b} \int_0^y y \frac{d\overline{U}}{dy}\, dy \qquad [11]$$

These two quantities are plotted in Fig. 3 and it is to be noted that the rate of transformation of pressure energy into turbulence energy in the entire center region is less than the rate of turbulence-energy dissipation. This again indicates that a large part of the energy of turbulence is created near the boundaries and diffuses into the main portion of the stream.

Another problem which it is interesting to analyze in the light of the ideas relating to turbulence and energy dissipation is that of the prevention of high velocities in conduits having steep hydraulic gradients. This is of practical importance in the design of fishpasses and the design of closed conduits where the velocity must be reduced for some reason and the cross section of flow kept relatively large. Taking the case of a circular conduit, the total rate of energy dissipation per unit time is equal to $Q\gamma(h/L)$, where γ is the unit fluid weight, Q the discharge, and h/L the loss of pressure head per unit length of conduit. It has been shown previously that this total rate of energy dissipation can be calculated also by integrating over the cross section the rate of pressure-energy conversion per unit volume of fluid. Thus there is obtained the relation

$$Q\gamma(h/L) = 2\pi \int_0^r \tau \frac{d\overline{U}}{dy}\, y dy \qquad [12]$$

For rough conduits and fully developed turbulent flow the mean shear τ will be determined practically entirely by the characteristics of the turbulence, primarily by the value of the mixing coefficient ϵ, since $\tau = \rho\epsilon d\overline{U}/dy$. For a given cross section and given friction slope, if we desire to reduce the mean velocity, the total rate of energy dissipation must be reduced. Offhand, it might appear that, to reduce the rate of energy dissipation, the amount of turbulence should be reduced. This problem will be clearer if for the moment we replace the coefficient of turbulent diffusion, ϵ, by the viscosity coefficient and imagine we are dealing with viscous flow. Obviously, to reduce the velocity of viscous flow for a given cross section and pressure drop, the viscosity must be increased. Therefore, it is logical that, for the case of fully developed turbulent flow, the value of ϵ should be increased, thus obtaining more intense mixing. For a constant cross-sectional area, or depth of open channel and constant hydraulic gradient, the mean shear remains the same, therefore, any increase in ϵ must result in a decrease in $\frac{d\overline{U}}{dy}$. Since according to Equation [12] the rate of energy dissipation is proportional to ϵ and to $(d\overline{U}/dy)^2$, it is apparent that an increase in ϵ will result in a decrease in the mean velocity. The value of ϵ can be effectively increased by the installation of specially designed obstacles along the conduit walls so as to produce discontinuities in the flow which will result in the creation of large eddies, thus producing intense mixing between the liquid at the walls and that in the center of the conduit.

Referring to Equations [6] and [7], which give the rate of turbulence-energy dissipation, it is noted that this rate is decreased by increasing the scale of the turbulence and by decreasing the velocity fluctuations. Of course, the rate of energy dissipation as given by Equation [12] must be the same as the rate of dissipation of the energy of turbulence by viscosity. However, since the turbulence is not isotropic, all that can be said is that the expression for isotropic-turbulence dissipation indicates the manner in which the various parameters describing the turbulence affect the energy dissipation. It is thus apparent that to decrease the rate of energy dissipation, which in the problem under consideration, means to reduce the mean velocity, eddies of large scale instead of large vorticity should be created.

Beyond bends, valves, transition sections, etc., a considerable amount of the energy producing flow is converted into energy of turbulence, which gradually dissipates downstream. This turbulence accounts for a major part of the pressure loss produced by any of the aforementioned apparatus. Ordinarily this energy of turbulence causes no other particular difficulty; however, there are some instances where it may create additional troubles. If a stream of water containing a considerable amount of turbulence energy is used for some purpose such as in a diffuser to convert part of the velocity energy into pressure energy, inefficient conversion will result. It has been shown, for instance, that the operation of various water-using equipment, such as closet-bowls and other plumbing equipment, can be improved if valves and other flow-disturbing apparatus are located sufficiently far away from the water-using apparatus so that the turbulence energy created has time to dissipate.

Solid streams of water passing through the atmosphere must be free of large intense turbulence eddies, otherwise the stream will spread and not remain solid. The importance of this particular item was demonstrated by Quick (7) in regard to the efficiency of impulse turbines. Fittings, valves, etc., upstream from the turbine nozzle introduce excess turbulence and, unless this turbulence energy is dissipated, it will appear in the jet issuing from the nozzle, causing spreading of the jet and decreasing the power it supplies to the wheel. In this connection, an item of importance is that, in a contracting stream such as in a nozzle, the longitudinal components of the turbulence are reduced considerably, while the transverse components are not, and may in fact increase (8).

In tests of apparatus such as valves, diffusers, bends, etc., varying the degree of turbulence in the approaching fluid is many times desirable and of interest, especially if flow separation occurs. It has been demonstrated that in many flow phenomena an increase in the energy of turbulence produces effects quite similar to an increase in Reynolds number. This fact has been applied in studying the transition of the boundary layer from viscous to turbulent flow in wind-tunnel experiments. An increase in Reynolds' number or an increase in the turbulence causes the transition to occur nearer the leading edge.

Kerr in his discussion of Quick's paper (7) on impulse turbines brings out the interesting fact that he was able to correlate field and laboratory tests better by introducing artificial turbulence in the approach conduits. The intensity of the turbulence is the parameter that has the most influence in altering flow characteristics past solid boundaries, however, Taylor (9) has shown that the scale factor of the turbulence does have some influence. An increase in the scale factor produces an effect opposite to that produced by an increase in the intensity. However, the relative effect of a change in scale is much less than the effect of a change in the intensity.

In general, preliminary investigations seem to indicate that the effect of turbulence in the approaching fluid on the energy loss through any flow-control apparatus or device is far from negligible. Turbulence is another property of the flow which must be controlled and its effect understood, if accurate correlation of test data, especially model-test results, is to be obtained.

Energy of Turbulence in a Diverging Conduit

Most of the energy lost in diverging conduits of greater total angle than about 4 deg is lost because of the creation of an extra amount of turbulence. Part of the kinetic energy of mean flow in the smaller conduit is converted into turbulence energy which is gradually dissipated downstream. Gibson (10), Peters (11), and others have shown that in general the efficiency of the conversion of the kinetic energy into pressure energy is least for a total angle of divergence of some 60 deg and that, at an angle of 40 deg, the efficiency of conversion is about identical to that at 180 deg, or sudden expansion. Depending upon the angle of divergence and the Reynolds number, separation may occur with the resultant backflow along the diverging boundary.

A study has been made by the author under the sponsorship of the American Society of Civil Engineers' Hydraulic Research

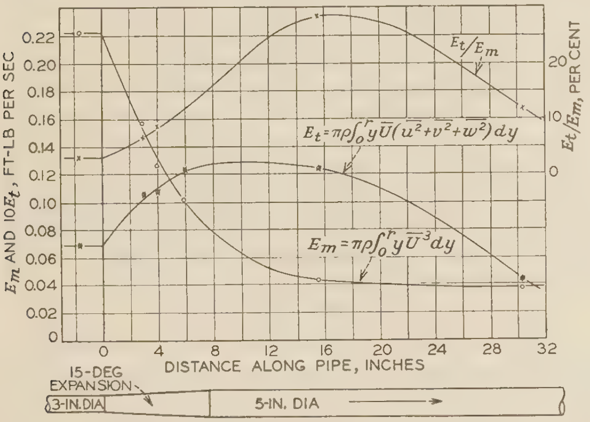

FIG. 4 VARIATION OF KINETIC ENERGY OF MEAN FLOW AND TURBULENCE ENERGY, ALONG A CONICAL DIVERGING CONDUIT

Committee relating to the mechanism of the energy changes that take place in diverging conical conduits. By use of a special technique (12), streaks formed by immiscible particles suspended in the water were obtained on motion-picture film. From the length and direction of these streaks, the longitudinal and transverse components of the velocity vector at various points in the fluid could be determined; from these it was possible to calculate at any point the mean velocity $\bar{U}$ and the parameters $\sqrt{\overline{u^2}}$ and $\sqrt{\overline{v^2}}$. The total kinetic energy of mean flow and the total turbulence energy, as given by Equations [1] and [3], could then be calculated at various sections in the expansion and beyond it. In calculating the turbulence energy, the transverse component v was assumed equal to w; this should be quite true in the central portion of the conduit, however, it does not hold near the boundary.

In Fig. 4 are shown typical data obtained on the energy changes in a 15-deg (total angle) expansion from a 3-in. to a 5-in. pipe. Though the absolute magnitude of the energy of turbulence may not be quite exact, nevertheless, its variation should be correct.

Note that the value of the energy of turbulence increases until the end of the expansion, remains practically constant for some distance, and then gradually decreases. Data were not obtained a sufficient distance downstream to indicate that normal conditions had been established, however, it is to be noted that the value of the total kinetic energy of mean flow was not changing very much after a distance of 2 ft beyond the start of the expansion.

The discharge for the data shown in Fig. 4 was 0.082 cfs, which gave a velocity of 1.67 fps in the 3-in. and 0.6 fps in the 5-in. conduit. No separation of flow appeared to occur in the expansion, probably due to the low Reynolds number. It is to be noted that, in the expansion and for some distance in the large conduit, the value of E_m is considerably greater than the value of the total kinetic energy of mean flow computed, using the average velocity in the cross section, which will be referred to as E_m' and is equal to $Q\gamma\bar{U}^2/2g$. In fact, just at the end of the expansion, the value of E_m is almost three times as great as the value of E_m'. At a distance of 2.5 ft beyond the start of the expansion, the ratio E_m/E_m' is still 1.3.

It is of interest to note that the total turbulence energy reaches a maximum value of 28 per cent of the kinetic energy of mean flow at a distance of about 10 in. beyond the end of the expansion, thus indicating that the relative intensity of the turbulence is greatest at that point. The ratio of E_t to E_m in the straight 3-in. conduit is only about 3 per cent.

The investigation outlined illustrates the possibility of making a study of the internal energy changes that take place in turbulent fluids when such fluids pass through apparatus causing a dissipation of energy due to the creation of an extra amount of energy of turbulence. Other flow-controlling and regulating equipment can be studied in a similar fashion. The acquisition of such data for various apparatus under different conditions would give the designing hydraulic engineer a better picture of exactly what is occurring at any point in the flowing fluid.

Regarding the study of flow in expanding conduits, mention should be made of the distribution of the turbulence energy in any cross section. In the very beginning of the expansion, most of the turbulence energy was concentrated in a layer a short distance from the wall. This seemed to be the place of greatest shear with the result that most of the excess turbulence was created in this layer. Farther downstream in the expansion, this region of maximum shear stress moved toward the center of the conduit, with a result that the turbulence became more uniformly distributed. The variation of shearing stress across a section of a diffuser has been studied by Schultz-Grunow (13), who showed that the maximum shear stress does occur in the interior of the fluid.

Delaying of separation in a diffuser, and thus causing a decrease in the energy of turbulence created, can be achieved to a certain extent by the inducing of spiral motion in the fluid entering a diffuser. Such spiral motion tends to cause the flow to follow the boundaries for higher mean velocities than it otherwise would. This was experimentally demonstrated by Peters (11).

Concerning the Origin of Turbulence

In practical hydraulics problems, the dissipation of potential energy or kinetic energy is, in the final analysis, a problem associated with the turbulence characteristics. In various hydraulic structures and apparatus, the shape of flow passages may be such as to produce regions of high internal shear, thus causing the creation of large amounts of energy of turbulence. A study of the origin and dissipation of this energy is possible and highly profitable. However, the main difficulty at present in any such study is the lack of efficient methods of measuring quantities such as the intensity and scale of the turbulence. To date the hot-wire anemometer has become practically a standard apparatus for obtaining turbulence measurements in air streams, however, its use in water has not been very successful. Photographic methods are being used in water and though the data obtained appear reliable and the necessary equipment is inexpensive, the time consumed in making analyses from motion pictures is somewhat excessive. The general method of obtaining the necessary data is to take motion pictures of thin color

streams or of immiscible droplets suspended in the flowing water.

There are many fundamental problems relating to the turbulence mechanism which merit study and such studies should help clear up many indefinite and obscure points. Probably one of the most interesting of these problems is that of the "origin of turbulence." In the regions where the turbulent flow is fully developed, it becomes difficult to obtain any idea of where and how the vortexes or eddies really originate. It appears that in hydraulically smooth conduits there is a sort of surface of discontinuity near the boundary, the "rolling up" of which results in vortex formation. This vortex formation at a boundary should take place according to a strict physical law. However, as these vortexes depart from the wall and enter the main part of the fluid, their individual behavior from then on is unpredictable. In rough conduits, if the roughness protuberances project up through the laminar boundary layer, each such projection must of course shed vortexes, probably in a fairly regular manner, and these are also dispersed into the main stream by the diffusive action of the turbulence.

In a uniform conduit, where an unchanging velocity distribution has been established, there is, of course, a statistical equilibrium between the creation of the vortexes and their dissipation in the main body of the fluid. At rounded entrances to smooth pipes, the formation of these vortexes at the boundaries is easily observable by the injection of a stream of color near the boundary. A sort of vortex trail is developed which gradually breaks up and disperses downstream, thus establishing normal turbulent flow.

Schiller (14) has made a very significant application of this observed phenomenon. He noted that the resistance to flow in the region of fully developed turbulence was essentially the same as in the region of the vortex trail near the entrance. From the arrangement and intensity of the vortexes, as revealed from a photographic study, it was possible to calculate their energy and dissipation. The dissipation of energy calculated using the vortex data agreed fairly well with the energy dissipation calculated from the measured resistance. Thus, the conclusion is reached that the dynamics of a turbulent flow can be calculated from the vortex trail from which it develops. Perhaps it will be studies of this character which will reveal the true physical nature of turbulence and make possible a better quantitative description of its characteristics.

Though hydraulic engineers may not be directly interested in such fundamental investigations regarding the dynamics of turbulence, nevertheless it should appear quite evident that such information has much practical value. In general, any future increase in the efficiencies of turbines, pumps, various fluid-flow-control apparatus, diffusers, etc., will necessarily be of relatively small magnitude. However, to obtain such increases it will be necessary to study and understand the inner mechanics of turbulent fluid flow more thoroughly. Though the problems are extremely complex, the advances made in the last few years are promising, and it is hoped that with increasing interest in this subject our knowledge will develop rapidly.

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Discussion

B. A. Bakhmeteff.[3] It would seem that future progress in practical hydraulics largely depends upon the understanding and mastery of "turbulence." Until recently, knowledge of turbulence has been most rudimentary and then grossly empirical. One is greatly indebted therefore to the author for his efforts to present the latest attainments of the theory in their possible relation to hydraulical problems.

Contributions of the type, illustrated in Fig. 4 of the paper and revealing the inside-energy course in an expanding conduit, are particularly welcome. Detailed quantitative observations of the inner mechanism of turbulence are extremely difficult and for that reason more than scarce. The author has perfected a novel technique and with its use has disclosed important and valuable facts. It is earnestly hoped that the author will persevere in his experimental explorations because the gathering of systematic factual material which, in an unmistakable manner, would elucidate the physical aspects of the turbulent process in all its stages, is the most important and urgent step for research to fulfill.

Present-day turbulent theory, in itself a remarkable and daring mathematical venture, in many respects has outdistanced direct experimental knowledge of turbulent phenomena. Many of the theoretical concepts and premises are still in a hypothetical and contestable stage. Indeed nothing could benefit and spur theoretical advance more than a solid background of indisputable facts.

The necessity for probing theoretical concepts is not limited to the latest advances of turbulent theory. Indeed the gulf between mathematical analysis and physical reality has been a lifelong feature in the realm of hydromechanics. A surprising example is offered by the traditional chain of reasoning dealing with dissipation of energy in viscous flow, the concluding link of which is presented by Equation [5] of the paper. The general treatment, typified by the presentation in Lamb's classical treatise,[4] has

[3] Professor of Civil Engineering, Columbia University, New York, N. Y. Mem. A.S.M.E.

[4] Bibliography (5) of paper, paragraph 329, p. 579.

never been questioned. The fact is, nevertheless, that when probed against the simplest practical cases, such as uniform laminar motion in pipes or between parallel plates, the reasoning leads to most unexpected and baffling conclusions, which only indicate that the initial premises underlying the analysis require attentive and constructive scrutiny.

The writer is especially interested in the closing part of the paper, dealing with the "origin of turbulence." He recalls discussing this rather evasive and controversial subject with the author and is happy to subscribe and support the views adhered to by him. From a practical point of view, two basic forms in which turbulent energy is engendered and subsequently dissipated, should be differentiated. The one, featured by Fig. 4 of the paper is typical of so-called "local hydraulic resistances" and can be appropriately termed "episodic turbulence," in contrast to "established turbulence" as typified by uniform pipe or channel motion.

"Episodic turbulence," being the source of so called "local losses," arises from concentrated turbulence-engendering activities which accompany brusque changes of flow forms (retardation or deflection). The process is especially enhanced by "separation." In fact both observation and theory[5] are in substantial accord regarding the way by which eddies are generated in the "free" discontinuity (separation) sheets inside fluids, as in the instance of submerged efflux from an orifice or in the wakes back of immersed blunt solids. The eddy-engendering activity is ascribed to the inherent instability of such "free" discontinuity sheets subjected to excessive stress. Similar to columns or thin plates, which buckle under excessive load, these separation sheets cease to conserve stability of form and, as the author points out, curl up into regular chains of individual eddies. The next stage is for these eddies to be "cast off" into the neighboring streaming, where they swarm in an unpredictable and random fashion, such irregular eddy motion constituting the physical essence of turbulence.

No direct visual testimony is available so far regarding the engendering of eddies near the walls of a smooth conduit in established turbulent flow. Indirect evidence amply indicates, nevertheless, that the process in essence may not be dissimilar to what takes place in a separation surface inside a fluid. The wall surface naturally restricts crosswise deformation. Hence a certain transversal latitude is required before the inherent instability, pertinent to the sheets of the highly stressed boundary layer, can assert itself. This transversal latitude is exemplified by the critical boundary-layer thickness $\delta_{cr} = (R\delta)_{cr} \times \frac{\nu}{U_b}$ (where ν = kinematic viscosity; U_b = border velocity of stream), which qualifies "transition" from laminar flow to turbulent. The origin of turbulence in "established" flow reverts thus to the boundary-friction zone which, in the process of expansion, reaches the critical stage and assumes at the outskirts of the laminar sublayer the role of a special eddy-generating region. Cast off into the central parts of the streaming, these eddies in their gradual spreading impose an appropriate turbulent pattern on the flow as a whole.

Hunter Rouse.[6] Fluid turbulence and its accompanying problems have been approached, so to speak, by the process of successive approximation. Long ago it was considered sufficient to take the effects of turbulence into account by means of a single coefficient, empirically determined for the boundary conditions in question. With the advent of more rigorous thought, attention was given to the analysis of turbulent flow along flat plates and through uniform conduits on the basis of certain general concepts regarding the physical nature of the turbulence mechanism. Eventually, however, it was realized that this very nature of the phenomenon rendered its exact analysis impossible without the use of statistical methods. To hydraulic engineers, this realization brought little comfort, for the tedious routine of laboratory investigation and the apparent mathematical complexity of the resulting functions are not in themselves particularly attractive. Therefore, the author deserves the gratitude of the profession at large, not only for undertaking the first statistical analyses of turbulence to be performed in the hydraulic laboratory, but for summarizing and explaining in hydraulic phraseology the essentials of this most recent approach.

Hydraulic engineers are wont to distinguish generally between two sorts of "turbulent" motion, considering the local disturbances produced by boundary changes as quite unrelated to the phenomenon ordinarily classed as turbulence in pipe flow. The author shows that any case of statistically irregular motion may be analyzed in terms of the same two types of parameter, the average eddy size and the average velocity of fluctuation which the eddies produce, one form of turbulence differing from another only to the extent that these parameters are differently distributed throughout the flow. Whether the flow is uniform or nonuniform, therefore, and whether the problem is one of energy loss, heat transfer, or sediment transportation, knowledge of the variation of these two parameters provides the key to the ultimate solution.

Were all turbulence isotropic in nature, both the experimental and analytical phases of the general problem would be relatively simple, for at any point the length and velocity scales of isotropic turbulence are the same in all directions. Unfortunately, most of the phenomena confronting the hydraulic engineer display definitely anisotropic characteristics. Not only are the three root-mean-square components of the velocity of fluctuation generally different, but a generally dissimilar difference is found in the mean dimensions of the eddies in the corresponding directions. A complete description of the turbulence pattern may therefore be obtained only through the determination of the velocity and length parameters in each of the three coordinate directions throughout the region of flow under investigation. This situation represents the principal obstacle to experimental progress, for no means yet exists of measuring more than two velocity components simultaneously, and the evaluation of the eddy dimensions remains dependent upon measurements of the velocity correlation at different points as functions of distance and direction.

In diffusion problems, to be sure, only the two transverse parameters seem to be of importance, as in the case of heat transfer or sediment suspension. On the other hand, only the longitudinal characteristics of eddy velocity and size can be measured with a single instrument. For instance, the revolutions of a current meter (provided that it is sufficiently small in proportion to the flow section) may be recorded in such a way as to provide from a single graph both the longitudinal root-mean-square fluctuation and the longitudinal mean eddy size. That is, the variation in amplitude of the velocity graph will yield the characteristic magnitude of the fluctuation, while the velocity variation with time (i.e., the frequency of fluctuation) should permit evaluation of a correlation coefficient R_x, somewhat similar to the factor R_y used by the author. Were it possible to establish the relationship between these characteristics and those for other directions, the measurement of anisotropic turbulence would thereby be greatly simplified.

The author discusses two different parameters for the linear scale of turbulence, the mean eddy size L and the "small" eddy size λ. The writer believes that the fact should be stressed that these are not strictly comparable length parameters, for the latter represents an arbitrary intercept of the horizontal axis of the cor-

[5] "Instability of Vortex Sheets," by L. Rosenhead, Proceedings of the Royal Society of London, vol. 134 A, 1931, p. 170.

[6] Professor of Hydraulics, State University of Iowa, and Consulting Engineer, Iowa Institute of Hydraulic Research, Iowa City, Iowa.

relation curve, while the former usually represents the area under the curve. That some types of turbulence apparently do have several different length (and velocity) scales of comparable type, however, is indicated by current-meter records of river flow in which high-frequency low-amplitude variation seems to be superposed upon a primary curve of lower frequency and higher amplitude; indeed, fluctuations of still lower frequency are evidenced by long, roughly sinusoidal waves in the velocity record, although variations of such great period probably do not merit the name of turbulence. Since, as pointed out by the author, the smaller eddies govern the rate of energy dissipation and the larger the diffusion characteristics of the turbulence, the writer raises a question as to whether the customary evaluation of the velocity record as a whole properly takes into account the relative magnitudes of these different turbulence scales.

Author's Closure

In order to bring the newer concepts of turbulent fluid motion into the realm of useful knowledge of the hydraulic engineer it is necessary to present accurate and clear physical interpretations. This is not always easy. Professor Bakhmeteff in his discussion helps to supply such interpretations which do much to further elucidate the various phenomena, particularly that of the origin of turbulence. The process and mechanism by which the energy producing normal turbulent flow in a simple straight conduit is transformed into heat is unbelievably complex. A few years ago we were satisfied to state that hydraulic friction consisted of the "rubbing together of fluid particles against each other and on the walls of the conduit." How entirely inadequate this description is has been impressed on those who have given this problem some thought and study in the light of our increased knowledge of the turbulence mechanism.

Analyses and observations indicate that the energy-producing flow is to a large extent transferred into the "eddy-generating" zone near the walls. In this zone the energy is transformed into turbulence energy of the rotating eddies which diffuse throughout the stream and gradually are dissipated. The mixing action of the eddies in the main portion of the stream is the mechanism by which the energy producing flow is transferred to the boundary zone. The process of transferring the potential energy producing flow from one region to another and the process of dissipation of this energy by the eddies after they are created occur, of course, in established turbulent flow, simultaneously; nevertheless, they are distinct and can be separated in any analytical study.

As mentioned by Professor Bakhmeteff even the mechanism of energy dissipation in viscous flow in conduits has not been perhaps adequately interpreted from the physical standpoint. Though the fundamental equations given in the classical hydrodynamic treatises are correct, they, however, do not clearly indicate what actually occurs, and thus may lead to faulty physical interpretations.

The goal of modern fluid mechanics is to probe more deeply into all fluid-flow phenomena, using the available theoretical concepts as guides and always depending on the laboratory to provide the necessary facts. Since a proper physical picture appears essential, the author has favored various photographic techniques in making the necessary experimental studies, even though much time is required to obtain the final data and facts.

Professor Rouse mentions that most of the turbulence phenomena of concern to the hydraulic engineer are not of the simple isotropic type. This is, of course, quite true, and therefore care must be taken in any attempt to extend the results of analyses and experiments dealing solely with isotropic turbulence. In general only the ideas and methods of attack can be borrowed, not the conclusions. Since no method exists at present for recording simultaneously the three velocity components in turbulent water, the best that can be done is to measure two components simultaneously, and another two in the other plane. Such measurements will give adequate data for obtaining the intensity and length characteristics of the turbulence in the three coordinate directions.

Professor Rouse's comments regarding the two different length parameters, which the author discusses in the paper, are quite pertinent. It is probably true that the relative magnitude of λ and L has no particular significance. Nevertheless, the particular definition of these length terms makes it possible to compare them for different conditions and to make conclusions regarding the energy dissipation and diffusion characteristics of different turbulence phenomena.

The encouraging comments of the discussers is appreciated and will prove very helpful in further work in this most interesting field of fluid mechanics.

Progress in Design and Performance of Modern Large Steam Turbines for Generator Drive

By G. B. Warren,[1] Schenectady, N. Y.

The first portion of this paper reviews briefly the progress of turbine and power-plant design over the last 20 years.

The second portion illustrates with numerous assembly drawings the progress of recent turbine design and shows both the standard and the special types of turbines being built to meet present requirements. In addition, it sets forth the general principles governing turbine construction and illustrates some of the new turbine constructions designed to withstand extremely high pressures and temperatures. The trend of turbine reliability and turbine economy is shown, and a number of test results of large modern turbines are made available.

The third section discusses detailed considerations of turbine design, and illustrates numerous turbine parts designed for the utilization of high pressures and high temperatures, but which may be of interest for other fields of application.

PART I—TRENDS IN POWER-PLANT AND TURBINE DESIGN

The progress of turbine design and construction since its inception has consisted largely in:

1 Building turbines and the attached generators of higher unit capacities to meet the growing demands for more power, and so permit reduction of the plant investment per unit of output.

2 Continuously modifying and refining the design, manufacturing processes, and materials used in order to increase the reliability of operation and to reduce the outage time which has always been one of the chief concerns of operators of such equipment.

3 Utilizing higher initial steam pressure and temperature, coupled with improved heat cycles, such as regenerative feed-water heating, resuperheating, air preheating, etc., to decrease the fuel consumption per unit of output.

4 Designing the turbine to utilize an ever-increasing proportion of the available energy in the steam cycle being used at the time.

The decrease of over-all plant investment through greater reliability of the turbine generator, and the decrease in investment and operating costs resulting from improved efficiency have always been of much greater significance to the owners of such equipment than reduction of first cost of the turbine and generator alone. The major endeavor, therefore, has been to improve these two factors through the use of new knowledge, new materials, and new tools and manufacturing processes. The principle of diminishing returns must be observed, however, and increased material and labor cannot be added out of proportion to the returns obtained, and every effort has been made to reduce costs where not inconsistent with the foregoing primary objectives.

With these forces at work, the turbine designers and builders have never been permitted to rest on their oars. Each year has seen new problems and new conditions to be met.

Hydrogen cooling[2] has made possible larger 3600-rpm generators, and has within the last few years permitted building many turbines for 3600 rpm rather than 1800 rpm.

Much of the progress mentioned has been summarized in a paper by W. E. Blowney,[3] but in order still further to illustrate this situation, Fig. 1 brings up to date a number of curves first published by Ernest L. Robinson[4] before the World Power Conference in Japan in 1929. The new figures show the average steam conditions for the turbines sold by General Electric each year, while the original curves showed but the maximum conditions. A better conception is thus gained of the trends.

Several conclusions can be drawn from these curves:

(*a*) Although the largest possible unit at any given speed has steadily increased, and the large units have been effectively utilized in reducing operating and fixed costs on the large utility systems, the average size of unit sold in the United States (units above 10,000 kw only considered) has increased but slowly.

(*b*) Increases in the maximum pressure and temperature have been followed by a steady rise in the average pressure and temperature for which machines have been sold, thus certifying to the fact that the pioneering done by a few power-plant owners has been followed by a general advance for the entire industry.

(*c*) The foregoing is further attested to by the almost continuous downward trend of the fuel consumption both for the best plants available and the somewhat similar trend for the average. Although it would appear from this latter curve that the continued progress in the average coal consumption may have slowed up, a closer examination of all the facts will serve to indicate that this is not the case.

The reduction in the coal consumption of the best plants is

[1] Designing Engineer, Turbine Engineering Department, General Electric Company. Mem. A.S.M.E.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

[2] "The Hydrogen-Cooled Turbine Generators," by D. S. Snell, Trans. A.I.E.E., vol. 69, 1940, pp. 35–50.

"Hydrogen-Cooled Turbine Generators," by M. D. Ross and C. C. Sterrett, Trans. A.I.E.E., vol. 59, 1930, pp. 11–17.

"Hydrogen Cooled Generators," by E. H. Freiburghouse and D. S. Snell, *Power*, vol. 82, no. 8, 1938, pp. 38–41.

"Hydrogen as a Cooling Medium for Electrical Machines," by Edgar Knowlton, Chester Rice, and E. H. Freiburghouse, Trans. A.I.E.E., vol. 44, 1925, pp. 922–934.

"The Application of Hydrogen-Cooling to Turbine Generators," by M. D. Ross, Trans. A.I.E.E., vol. 50, 1931, pp. 381–386.

"Liquid Film Seal for Hydrogen-Cooled Machines," by C. W. Rice, *General Electric Review*, vol. 30, 1927, pp. 516–530.

"Hydrogen Cooling of Rotating Machines," by C. M. Lafoon, Trans. A.I.E.E., vol. 55, 1936, pp. 703–709.

[3] "Turbine Trends," by W. E. Blowney, *Power*, vol. 83, no. 1, 1939, pp. 74–76.

[4] "General Trend of Steam Turbine Development by the General Electric Company," Trans. Tokio Sectional Meeting, World Power Conference, Tokio, Japan, 1929, vol. 3, pp. 1066–1078.

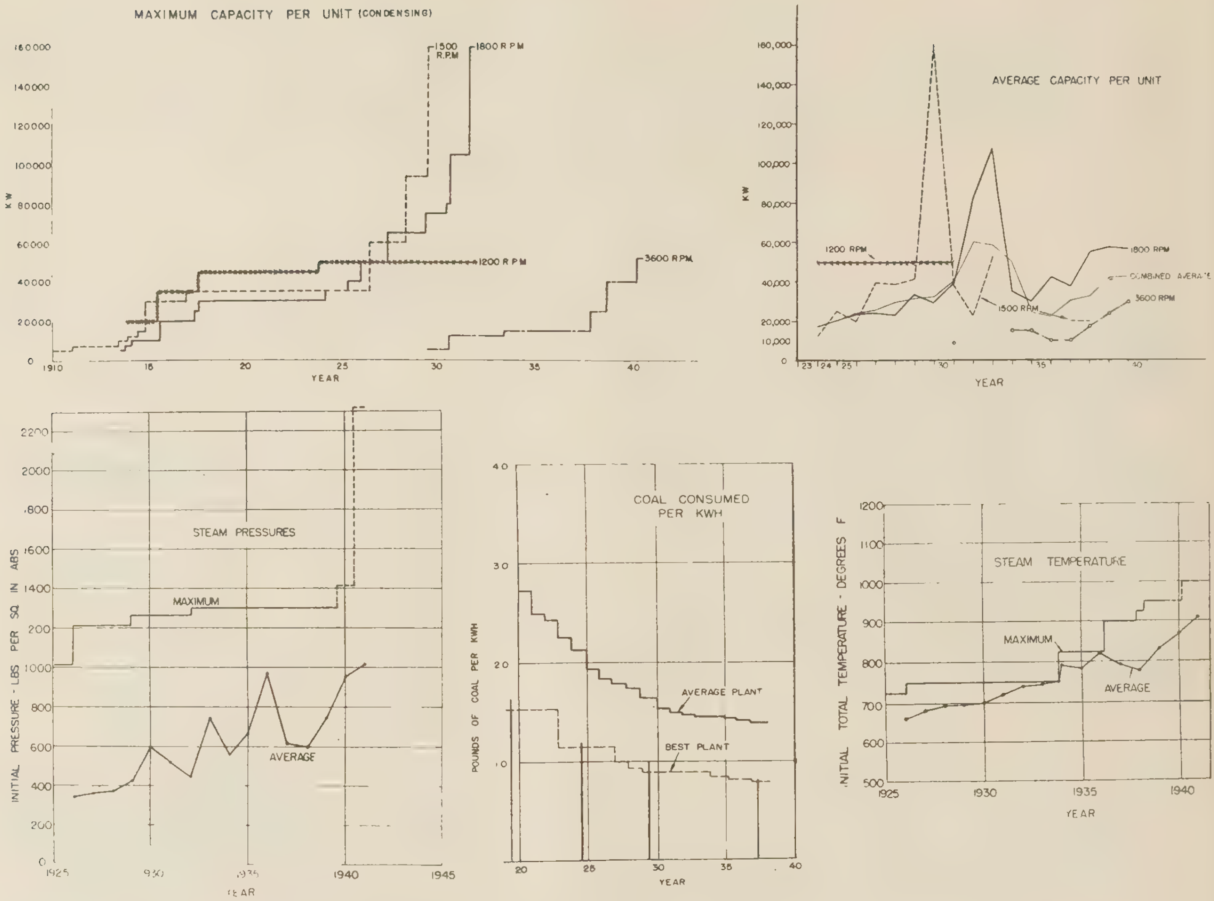

Fig. 1 Curves Summarizing Statistics of Turbine Progress
(Maximum capacity refers to single-shaft units. Larger 1800-rpm cross-compound units have been built up to 208,000 kw.)

due to the progress already noted which has continued during the last few years; whereas, the continual reduction in the average is due to the taking over of the load by newer and more efficient stations as they are installed from year to year, and to the relegating of the older and less efficient plants to peak-load and stand-by service. The last several years of business depression have served to reduce the number of new plants being installed, although the growth of the electrical load has continued. This resulted in an increasing use of many older and less efficient plants with a slowing up of the year-by-year reductions in the average coal consumption per kilowatthour. With the resumption of the power-plant building program which is now under way, we should see still further reductions in the coal consumption per unit of output. This is borne out by the latest preliminary figure of 1.37 lb of coal per kilowatthour recently published[5] for year 1939.

Turbine Speeds

The trend toward higher turbine rotative speeds has been dictated by the requirements of efficiency, reliability, and cost.

Where the leaving loss from the last stages of condensing turbines is not a determining factor, it is usually true that the turbine efficiency ratio, i.e., the relation between the energy which it makes available as output and that theoretically available, is increased as the design speed is increased. In that part of a turbine with a high volume flow, i.e., on most low-initial-pressure turbines and on the condensing end of all turbines, this increase is small, and may even be negative. On the other hand, when the volume flow is low, as on the high-pressure end of even large-capacity turbines, and on almost all small turbines, the gain with increasing speed is greater.

Of even greater importance is the fact that it becomes very much easier to build reliable turbines for high pressures and temperatures if the dimensions of the shells and rotors and interstage diaphragms are kept as small as possible. This can be done in the higher-speed turbines. As an extreme example, it is doubtful if it would be practical to design a 1200-rpm turbine of reasonable efficiency to operate at 1200 psi pressure, 950 F; whereas, such turbines are now being built for 3600 rpm with ease.

As an illustration of this situation, Figs. 2 and 3 have been prepared showing comparable turbine rotors and turbine casings for 50,000 kw condensing turbines at 1200 rpm, 1800 rpm, and 3600 rpm, respectively. These designs were for steam conditions of 275 psi, 650 F, 375 psi, 725 F, and 1200 psi, 900 F, respectively.

It is apparent that the requisite shell-wall thickness, etc., and increased shaft span that would be necessary to make the slower-speed machines good for the higher pressures and temperatures would be prohibitive. The deflections and distortions which would result, together with the great bearing span, would also prohibit the maintenance of the reasonably small clearances upon which high efficiency with high pressure depends. There seems to be reason to believe, supported by evidence on both

[5] Annual statistical number of the *Electrical World*, vol. 113, Jan. 13, 1940, p. 81 (105).

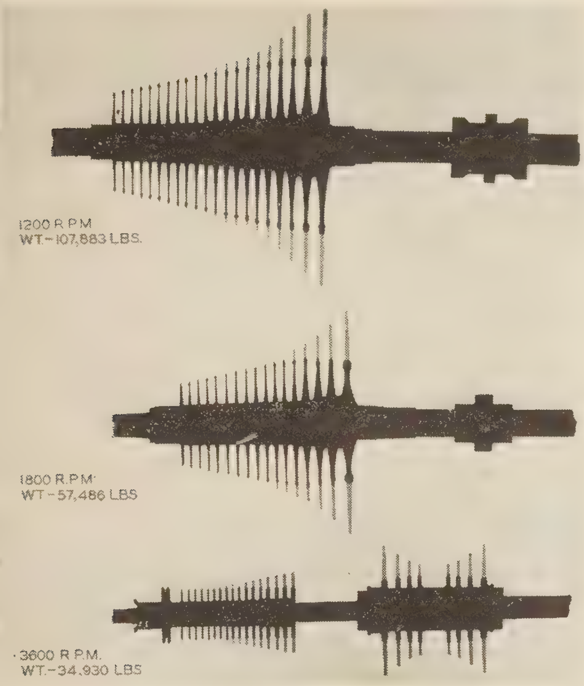

Fig. 2 Turbine Rotors for 50,000-Kw Condensing Turbines (1200 rpm, 275 psi gage; 1800 rpm, 375 psi gage; and 3600 rpm, 1200 psi gage, respectively; approximately to same scale.)

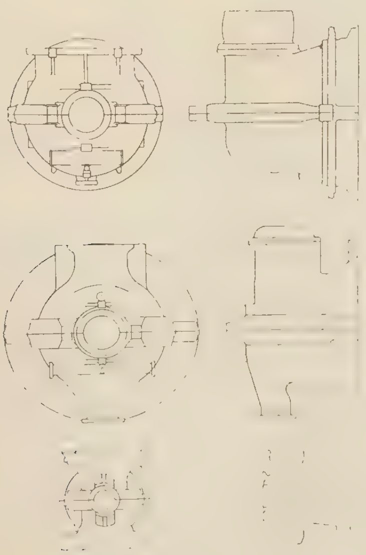

Fig. 3 Turbine Shells for 50,000-Kw Condensing Turbines (1200 rpm, 275 psi gage; 1800 rpm, 375 psi gage; and 3600 rpm, 1200 psi gage, respectively; approximately to same scale.)

land and marine turbines,[6] that the newer high-speed turbines are more reliable as well as more economical than the older slower-speed machines. This is contrary to the preconceived ideas of many engineers, and generally contrary to reciprocating-machinery experience. Nor do higher speeds generally mean higher stresses in the turbine elements. In almost every case the stresses in the new 3600-rpm turbines are lower than in the 1800-rpm machines of a few years ago.

The advantages of the higher speed have been particularly apparent in connection with high-pressure and high-temperature condensing, or so-called "topping," turbines.

The higher speeds have permitted better utilization of materials and should eventually reduce costs. The simultaneous increase in steam conditions, the utilization of better, more refined, and more costly materials as a means of increasing reliability, together with increasing labor costs as compared to several years ago, make relative cost comparisons between present and past practice difficult and often misleading. Costs are undoubtedly lower than they would have been had speeds not been increased. Turbine speeds for generator drive seem to have reached a limit at 3600 rpm with 60-cycle power being generated. Although new electrical devices may change this situation, it does not appear to be in the immediate future, and new development should be stabilized at this speed, which will probably be an advantage for all concerned.

PART II—GENERAL DESIGN AND PERFORMANCE

The recent trend of turbine development can best be illustrated by discussing specific turbines, somewhat in chronological order. The requirements of the various customers are still so diverse that each turbine is more or less a custom-built machine. This, however, has not been as costly as is oftentimes thought, because it has been possible in a large percentage of the turbines built to utilize more or less standardized portions of turbines with only minor modifications of these parts from machine to machine.

Although all of the turbines described in this paper bear a "family resemblance," there are still many differences. Actually, this is because in most cases there is not just one solution to a problem, but several. When one solution has been found to work, it may be incorporated in subsequent turbines built for closely similar conditions over a period of years.

In the meantime, turbine development may have been following along another solution, because this other solution may give promise of meeting future conditions better than the previous one which is still in use, and, hence, newer machines in process of design will embody the newer arrangement, although for certain current conditions of operation it may be no better than an older arrangement still also in use.

Fig. 4 shows cross sections of two modern turbines for rather moderate steam conditions, that is, 300 psi to 400 psi pressure and 700 F to 750 F temperature. The first is for 15,000 kw, 3600 rpm; and the second, for 75,000 kw, 1800 rpm. In both these machines steam at full boiler pressure and temperature is bypassed around early stages to obtain full load. Turbines of both these types have been fully tested and have, as will be shown in a later part of this paper, extremely high over-all efficiencies.

Fig. 5 shows four very recent machines designed for modern higher steam pressures and temperatures and for capacities ranging from 10,000 kw to 80,000 kw. The first machine shown is for 10,000 kw to 15,000 kw capacity and for 600 psi to 800 psi pressure, 825 F to 900 F temperature. The second machine is for 20,000 kw to 25,000 kw capacity for 800 psi, 900 F to 950 F temperature.

The third machine is typical of the 3600-rpm tandem compound turbines for from 30,000 kw to 50,000 kw capacity and for steam conditions varying from 600 psi to 1200 psi and temperatures from 825 F to 950 F. The fourth machine, typical of the large 1800-rpm single-casing turbines, is for from 75,000 kw to 85,000 kw capacity, and for steam conditions from 800 psi to 1250 psi, 825 F to 900 F. A turbine similar to this latter machine, rated

[6] "New Engineering in the Navy," by Charles Edison, Secretary of the Navy, *Scientific American*, vol. 162, March, 1940, p. 138.

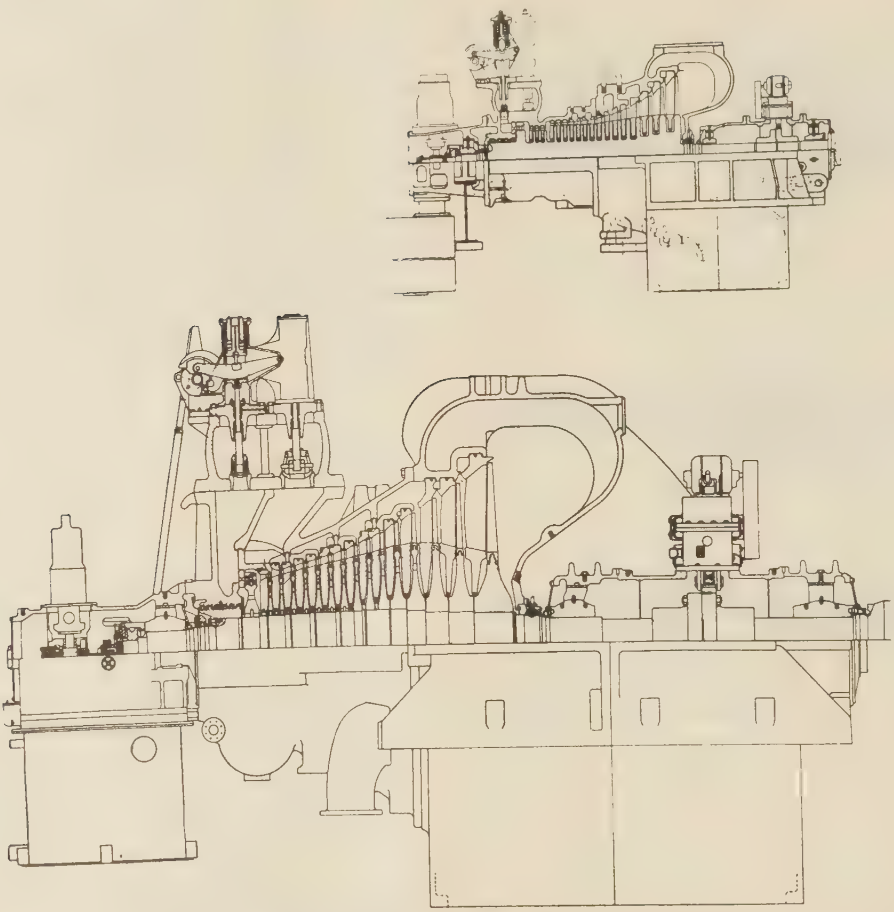

Fig. 4 Cross Sections of 15,000-Kw, 3600-Rpm, and 75,000-Kw, 1800-Rpm Condensing Turbines for 300 to 400 Psi, 750 F
(Approximately to same scale.)

80,000 kw and to operate at 1250 psi, 900 F, is now being installed for the Central New York Power Corporation.

Controlling Valves and Design of Early Stages

The general use of higher and more efficient steam conditions has had an important bearing upon the design of the early stages in turbines and upon the arrangement of the controlling-valve system used to permit the efficient meeting of a wide range of outputs from the turbine. In general, a two-row velocity stage first-stage wheel is the best design for a turbine with comparatively low volume flow, that is, somewhat under 50 cfs. This is because, although the two-row velocity stage wheel in itself is not as efficient as corresponding single-row impulse wheels, it gives a quick drop in the initial pressure and temperature through the first-stage nozzles, and reduces the pressure on the high-pressure packing and on the diaphragm packings, the losses through which are major items with a low volume flow turbine, and also reduces the rotation loss of the early stages. This arrangement also produces a relatively constant efficiency over a wide range of load.

If the volume flow is between 50 and 125 cfs, general experience indicates that the first stage should also be a two-row wheel as on the smaller flow machines, but that after approximately 75 per cent to 80 per cent full flow is obtained, full steam flow should be obtained by by-passing the first-stage wheel, and admitting full boiler pressure and temperature to the first-stage shell. This is sometimes called a single by-pass machine. On machines of larger volume flow than the foregoing, the two-row first-stage wheel, which is generally here full peripheral admission, may be retained, but designed so that at full admission to this wheel, about 60 per cent to 65 per cent of full-load flow is obtained. Eighty per cent of full load is obtained by admission of steam at full pressure and temperature to the first-stage shell; and full flow is obtained by by-passing steam at nearly full boiler pressure and temperature to a still lower stage. This is sometimes called a two by-pass machine. It is seldom desirable to carry this process further. The aforementioned volume flows are for 3600-rpm turbines. The corresponding flows for 1800-rpm turbines will be between three and four times these values. This latter construction is illustrated in Fig. 4. This by-passing method of regulating the flow is advantageous from many standpoints, but the author feels it is generally not so desirable when the pressure and temperature are high because of the larger portion of the turbine exposed to these conditions and the heating of the steam passing through the idle stages.

The method used on the smaller-volume-flow machines with a two-row velocity-stage wheel which is not by-passed is excellent for high pressures and temperatures, because it does not permit these steam conditions to go in their full intensity past the first-stage nozzles; to continue this system on the larger machines, however, would not result in the most efficient possible design for the lighter load conditions of such turbines. If

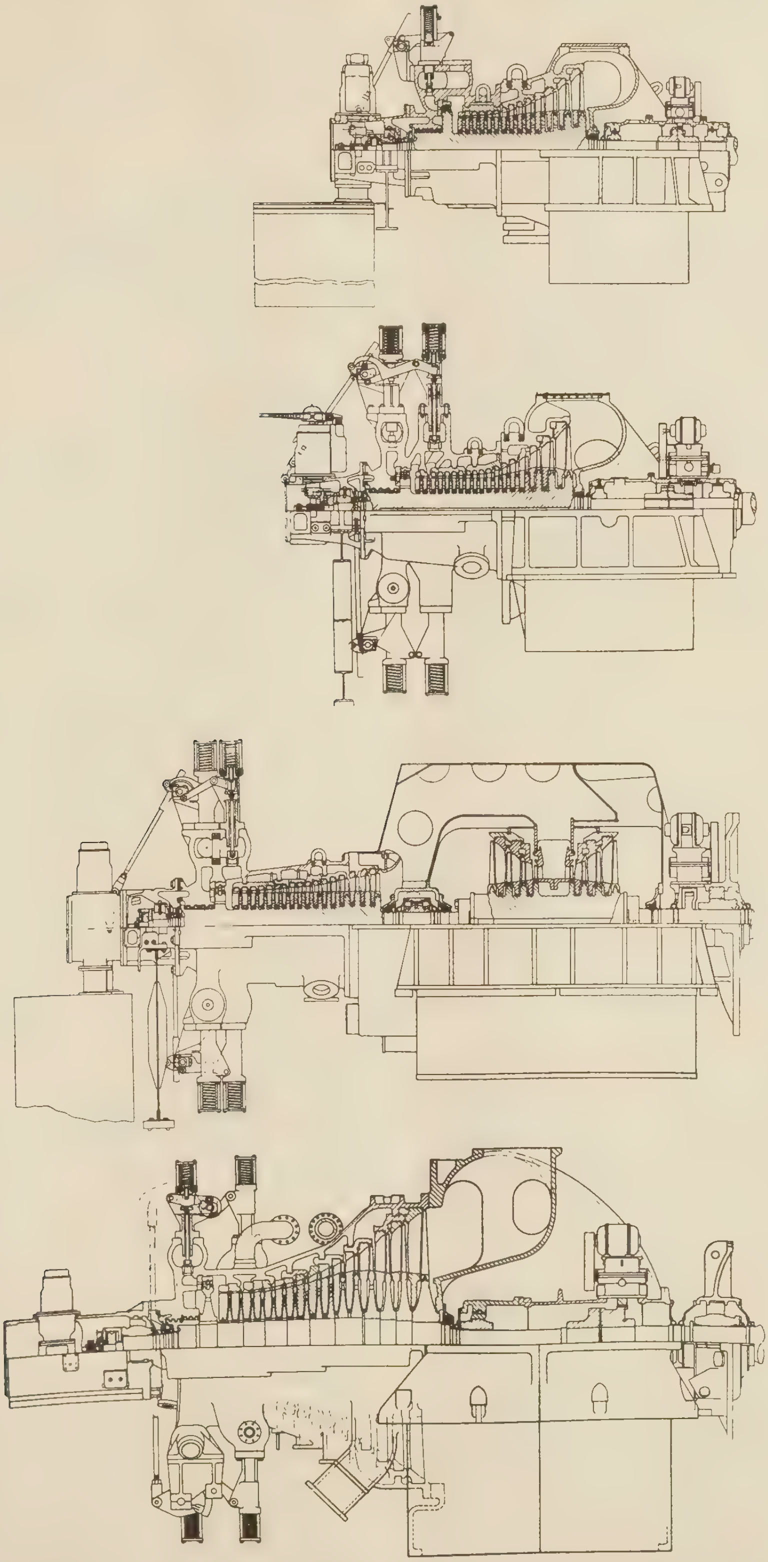

Fig. 5 Cross Sections of 10,000-Kw to 80,000-Kw Condensing Turbines for Modern Steam Conditions
(600 to 1200 psi and 825 F to 950 F; approximately to scale.)

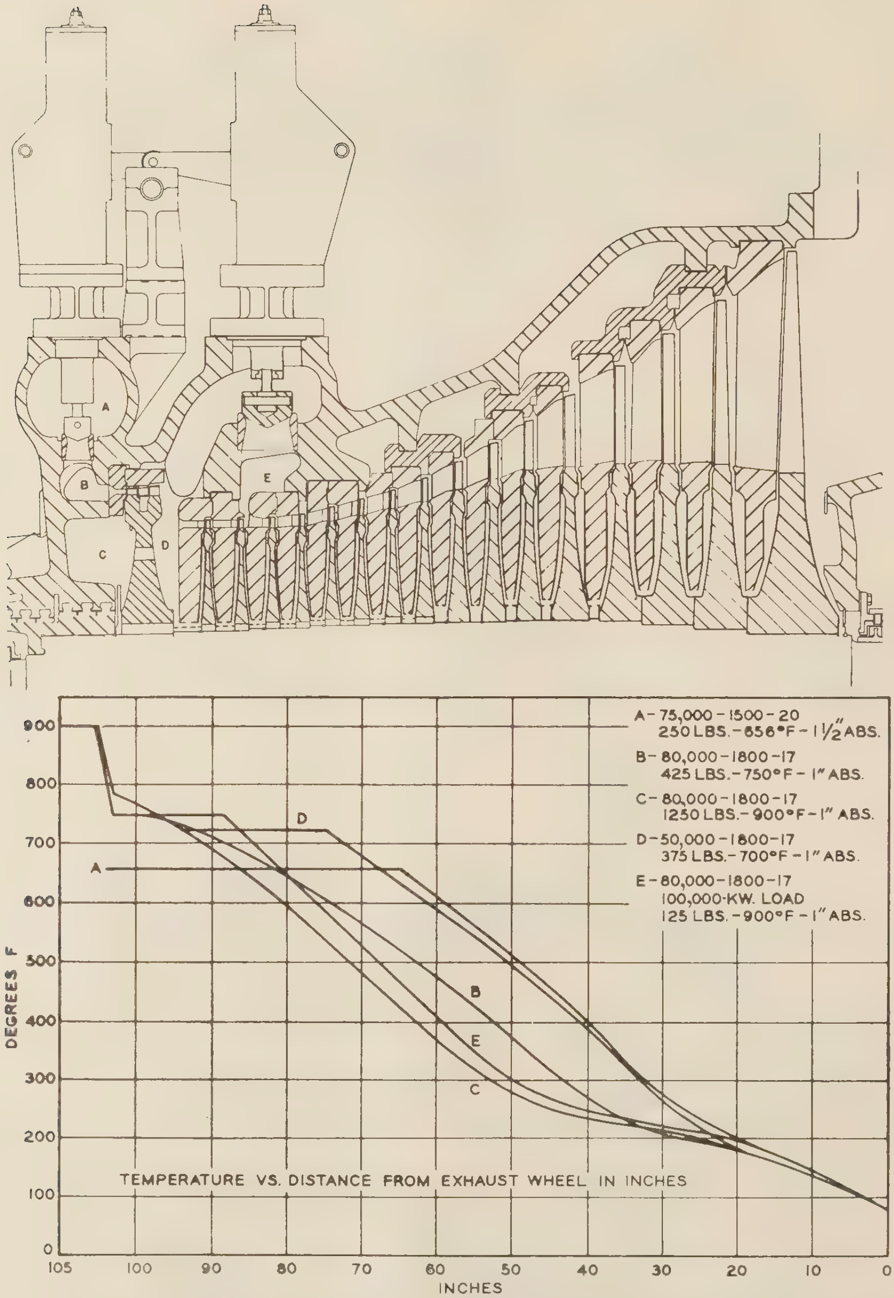

Fig. 6 Diagrammatic Cross Section of Internal By-Pass Valve Condensing Turbine and Variations of Temperature Through Turbines With External and Internal By-Pass Valves

(Curves *C* and *E* are turbines with internal, *A*, *B*, and *D* turbines with external, by-pass valves.)

the by-passing method of control just described is used, the shells and other turbine parts are subjected to the full intensity of the initial pressure and temperature of the steam. Therefore, an older construction, often referred to as an "internal by-pass valve," was revived and incorporated in the design. This permits a nearly equivalent result from the standpoint of economy, and at the same time protects the parts of the turbine against the incidence of these unusually high steam conditions.

A clearer view of the internal by-pass arrangement is shown in Fig. 6. The upper portion shows the arrangement on the 80,000-kw single-cylinder condensing turbine previously referred to. The lower portion shows a group of curves in which the maximum operating temperature at the various stages is plotted for a number of turbines rated from 50,000 kw to 80,000 kw capacity and with various steam-admission arrangements. Some of the curves represent the temperatures in older, lower-pressure, lower-temperature turbines with by-passing similar to those shown in Fig. 4. Curves *C* and *E* represent corresponding temperatures with the internal-by-pass machine with the by-pass both open and closed, and indicate that this construction permits the building of a single-cylinder turbine for 1200 psi, 900 F, which, with the exception of the first-stage nozzle and valve chest has

temperatures throughout no higher and generally less than older type turbines operating at much lower conditions. This type of design should result in a substantial increase in the temperatures for which turbines can be safely designed with present material, and should also result in decidedly less distortion and creep of turbine parts as a result of operation at these higher steam conditions.

Another refinement incorporated in some modern turbines is the placing of the controlling valves symmetrically in the top and bottom halves of the high-pressure shells. This tends to preserve the alignment and, hence, maintain the smaller clearances so necessary to high economy, despite the greater expansions incident to the present higher temperatures. This can be noted by contrasting the designs of the three larger machines in Fig. 5 with the designs shown in Fig. 4.

Design of Last-Stage Buckets and Exhaust Hoods

It is common knowledge that the projected area through the last-stage bucket annulus is generally the limiting factor in connection with the capacity for which condensing turbines can be built. It is possible within reasonable limits to build the earlier stages of any given turbine carrying a given last-stage bucket so as to pass any required quantity of flow, and to obtain any given capacity, providing, of course, the parts are made strong enough. Under such circumstances the velocity of steam leaving the last-stage buckets would go up approximately as the capacity, and the energy in the steam which is thus thrown away would go up as the square of the capacity. A point is soon reached at which the loss in energy from this so-called "leaving loss" or "exhaust loss" is so great as to make the increase in capacity obtainable this way wholly uneconomical. There is no hard and fast line which can be drawn, or which says that the leaving loss should be such and such a percentage in commercial machines. It generally ranges from 2 per cent to 6 per cent at maximum load, depending upon the steam conditions, vacuum, extraction, and the like.

Mechanically, there is a rather definite limit to the annulus area which can be obtained at any given speed, and at any given stress in the root of the buckets and their attachments. The actual value depends upon the skill and experience of the bucket designer, and upon the stress which his experience indicates he can allow safely in the materials available. This limit at the present time is approximately 25 sq ft of projected annulus area (pitch circumference times the length of bucket) at 3600 rpm, and about four times this value at 1800 rpm.

The kilowatt capacity which can be economically put into a turbine with a given last-stage bucket area depends upon the steam conditions. This capacity generally increases as the initial pressure and temperature are increased, if resuperheating is used, and if steam is extracted for regenerative feedwater heating, or for any other purpose.[7] This economical capacity is greater at poorer vacuums.

A rough rule would indicate that from 600 kw to 1200 kw maximum turbine capacity can be obtained economically from each square foot of last-stage bucket annulus, depending upon the conditions. Therefore, 80,000-kw to 100,000-kw, single-flow, 1800-rpm turbines can be built, but in 3600-rpm turbines it is generally necessary to "double-flow" above 25,000 kw capacity, and at the present time on a cross-compound, 100,000-kw, 3600-rpm turbine, described later, a quadruple flow exhaust to the condenser is being used.

The tip speed of these last-stage buckets is so great that the effect of the impingement of the tips of the buckets upon the slow-moving moisture particles in the steam causes erosion of the metal from the last-stage buckets.

Higher initial temperature or resuperheating reduces the moisture present, or the moisture can be separated by centrifugal force from the steam before it reaches the last stage. This latter is done, in so far as possible, by circumferential chambers surrounding the tips of the latter stages shown in the preceding turbine cross sections.

Experience has proved that these chambers are effective in removing the heavier moisture particles and in greatly reducing the erosion. The total effect on efficiency is not very great, ranging from about one quarter to three quarters of one per cent, being less on the larger and lower-speed machines.

Another method of protecting these buckets is to attach strips of harder material to the tips of the buckets. Stellite, as shown by the results of extensive research both here and abroad, seems to be the most effective material for preventing such erosion. Its superior qualities in this respect are all out of proportion to its hardness. These shields are generally attached to the outer portion of the higher-speed last-stage buckets.

The erosion of the next to the last-stage bucket does not appear to be a major factor on account of the higher density of the steam, and, hence, the higher absolute and lower relative velocity of the moisture particles before they are struck by the bucket edge.

Much research work in mechanical design and steam flow has been carried out to assist in the design of the last-stage buckets. However, much remains to be done. From both foregoing standpoints the design must be a compromise between (1) the varying stream-flow requirements at the root and at the tip, (2) the varying requirements of strength for the root and lightness for the tip, and (3) sufficient rigidity to the whole structure as to give the proper vibrational characteristics.

These latter-stage wheels and buckets must be tuned so that they will not vibrate at running speed,[8] but they would probably break if operated for very long under load at certain speeds either below or above this designed operating speed. In variable-speed marine turbines special precautions must be taken to see that the buckets are proportioned so as to be safe, at all speeds. This is not generally difficult for the sizes, speeds, and vacuums generally encountered.

Exhaust Hoods

The exhaust hood is the chamber into which the steam from the last-stage bucket of a turbine discharges. If this chamber is extremely large, no loss in pressure between the last-stage bucket and the entrance to the condenser will take place, but such a hood would be too costly, and would probably not be sufficiently rigid to hold the high-pressure portions of the turbine in alignment with the bearings. An exhaust hood which is too small would not permit the turbine to utilize the full condenser vacuum, and so would result in excessive losses.

A great deal of research, some of which was described by H. L. Wirt in 1924,[9] has been carried out by means of model tests in order to ascertain the best shape for building these exhaust hoods.

It is rather strange to find that under these circumstances the original designs, made more or less on the basis of judgment and instinct, are still frequently in use and are about as good as anything which the research work has disclosed. These

[7] "Recent and Possible Future Developments Affecting the Economics of Large Steam Turbine Practice in the U. S.," by G. B. Warren, Transactions of the Second World Power Conference, Berlin, 1930, vol. 5, pp. 107–141 and *General Electric Review*, vol. 33, 1930, pp. 434–443, 522–527.

[8] "The Protection of Steam-Turbine Disk Wheels From Axial Vibration," by Wilfred Campbell, Trans. A.S.M.E., vol. 46, 1924, pp. 31–60.

[9] "The Turbine Designers' Wind Tunnel," by H. L. Wirt, *Mechanical Engineering*, vol. 47, 1925, pp. 13–17.

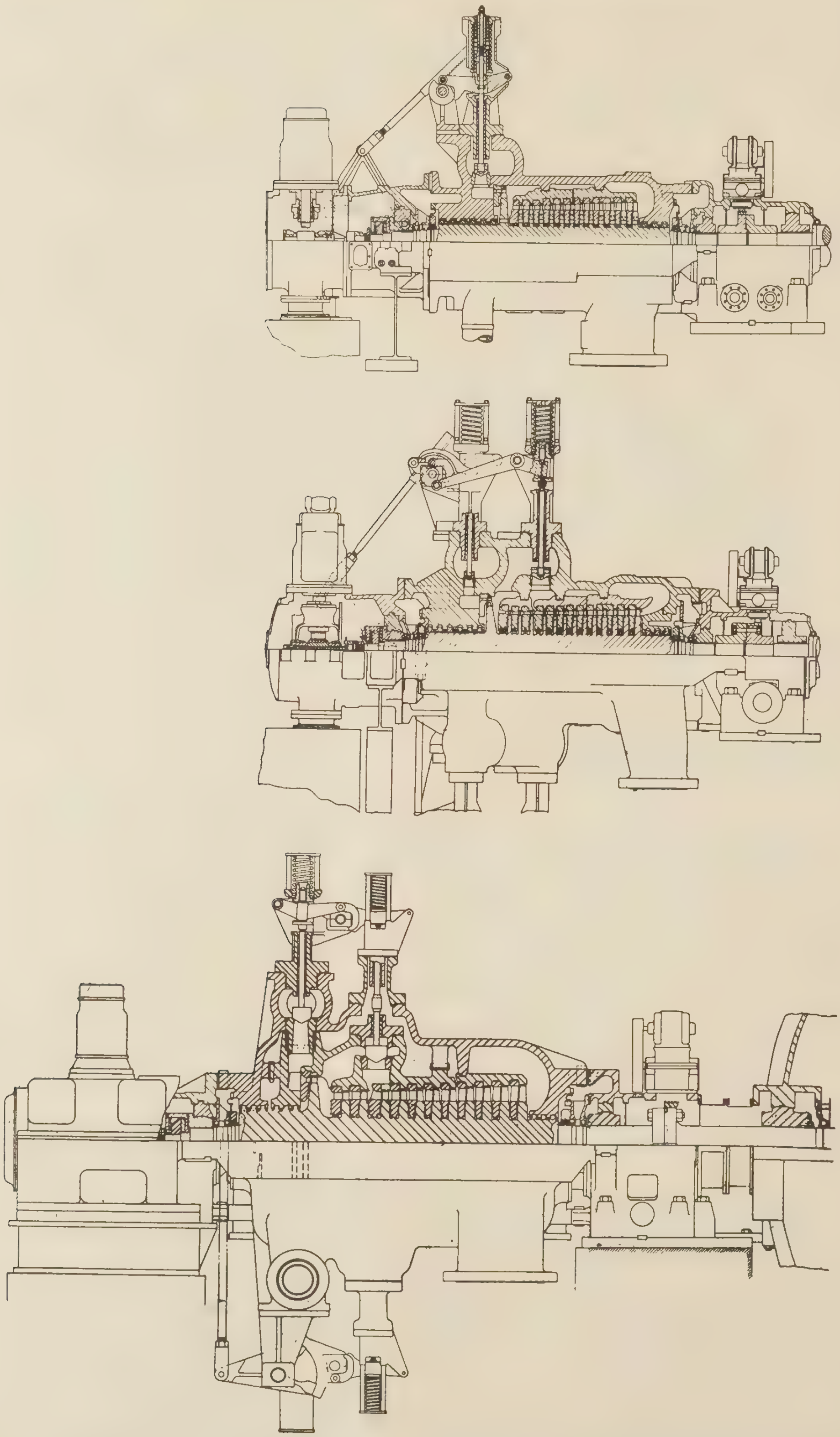

Fig. 7 Cross Sections Through Noncondensing Turbines for High Initial Pressures and Temperatures

(Approximately to scale.)

original designs, however, have been refined in certain ways as a result of this testing, and have been made much better from the mechanical standpoint as a result of operating experience with turbines in the interim.

Some turbines have been built where condenser placement on the turbine operating level has permitted a "stream flow" or "diffuser" type of exhaust hood. On these units vacuum measurements were obtained at the last wheel which were greater than the vacuums existing in the condenser. These hoods have been in service for a number of years on several turbines in the Chicago district with great success, but generally this type of condenser arrangement has not been favored by station designers on account of the extra space and unconventional arrangement required, although a gain somewhat in excess of one per cent results.

Noncondensing Turbines

Fig. 7 shows three sizes of noncondensing turbines for high pressures and temperatures. These have been built in quite a number of cases to supply industrial process steam, or to operate as superposed turbines exhausting into existing lower-pressure condensing turbines.

The first of these is for 10,000 kw to 15,000 kw capacity, 800 psi to 1200 psi pressure, and up to 900 F. This size machine generally requires no by-pass, and passes all of the steam through the first-stage wheel.

The second is a recently designed machine for 15,000 kw to 20,000 kw capacity, for the same pressure range as the first, and up to 925 F. In this size a by-pass becomes advisable in order to secure the best possible light-load economy. The design shown is one of the latest involving an internal by-pass, which, as previously pointed out, permits all of the steam to pass through the first-stage wheel at all loads.

The third shows a type of machine which has been built for from 25,000 kw to 60,000 kw capacity, for a wide range of steam conditions up to 2300 psi pressure and up to 950 F.

The first two of the foregoing machines are of the single-shell type. The third machine is of a type which was developed only a few years ago, and is known as the "double-shell" construction. It has been successfully applied to a number of installations at high pressures and temperatures.

Operating experience on seven of these turbines now in commercial service has been decidedly favorable. Trouble from steam leaks has not occurred. Shell distortions have been reduced to much less than ever previously experienced even in turbines at lower pressures and temperatures. The dismantling of this type has been made easier owing to greater ease of handling the smaller shell bolts and the reduced shell distortions. Several other machines of this type are now being installed or manufactured.

Development of Double Shell[10]

Double-shell construction has had a rather interesting development through four steps, the last of which is not yet in operation. The various steps in this design are shown in Fig. 8.

Basically, the principle is to surround the working parts of the turbine with a steamtight inner shell carrying its own bolting flange, and to build around this a second shell. The space between the two shells is maintained by communication with a lower stage in the turbine at a pressure intermediate between the intial pressure in the inner shell and the atmosphere. The total pressure drop is thus divided into two stages; neither shell

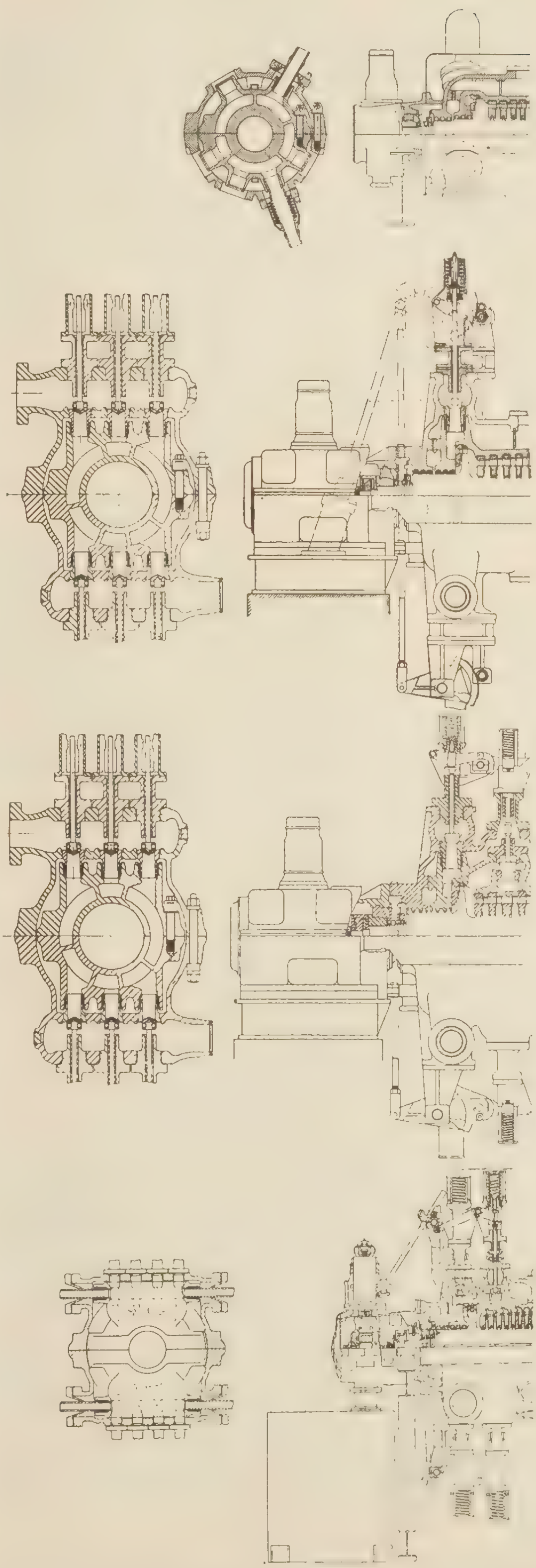

Fig. 8 Various Double-Shell Designs Showing Successive Developments

[10] "Logan Double-Shell Turbine," by G. B. Warren, *Power*, vol. 83, 1937, pp. 302–305.

"53,000-Kw 3600-Rpm Superposed Turbine for Waterside Station," by G. B. Warren, *Combustion*, vol. 9, 1938, pp. 27–32.

need be as thick as it would otherwise have to be in a single-shell machine. The inner shell is heated by steam on both sides, and as a result of all of these factors there is much less tendency for the shells to distort and the turbine may be much more rapidly brought up to operating temperature without the danger of undue internal stresses or distortions. Fig. 9 shows views of the horizontal bolting flanges of comparable single- and double-shell designs.

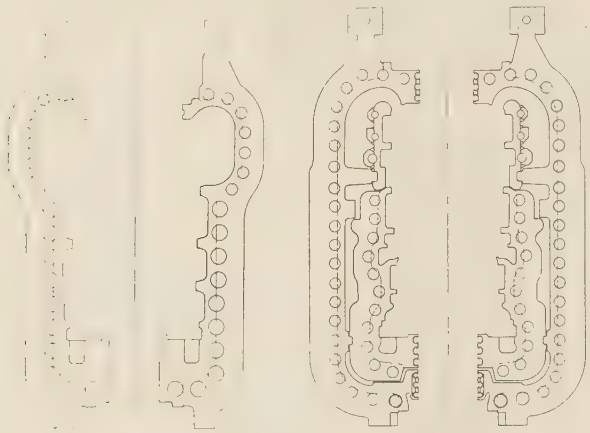

Fig. 9 Comparable Bolting Flange of Single- and Double-Shell Turbines

Type I in Fig. 8 is the first embodiment of this design, of which eight are built or building and five have now been put in operation. It has been quite successful but required an external controlling valve chest or chests and complicated piping.

Type II was the next design of which four were built or are building, and two have now been put in operation. This construction eliminated the external valve chest but preserved the advantages of the double-shell construction, embodying the valve chests in the upper- and lower-half outer shells or casings which made the turbine much more compact.

Type III shows the third embodiment of this construction in which to Type II an internal by-pass valve has been added. Seven machines of this type are being built. This design secures good economy over a wider range of load and at the same time permits a drop of pressure and temperature through the first-stage wheel and nozzle at all loads, with resultant protection to the turbine elements from the higher pressure and temperature of the inlet steam.

Type IV shows a still further development of this design in a condensing turbine to operate ultimately at 1250 psi gage pressure, 1000 F. In this last construction the entire upper and lower controlling valve chests are cast integral with the inner shell, the controlling valves operated by valve stems coupled to other valve stems coming through the outer shell, and thus the outer shell is protected at all points from the initial steam conditions and is only in contact with steam at much lower pressures and temperatures. The first-stage nozzle and the incoming steam pipes are the only parts of the turbine subjected to the full initial steam conditions. These parts are really such a small portion of the turbine that if need be they might be considered renewal parts, although it is not felt, with steam conditions now under consideration, with present-day materials, and with stresses which are now being run, that these parts will have to be renewed.

Interesting and Unique Turbines of Recent Design

The turbines shown in Figs. 4, 5, and 7 have been designed to fit more or less standard conditions, and they conform in general to the suggested standards issued in 1938 by the Federal Power Commission.[11]

In the last few years, however, a number of very interesting turbines have been designed to meet necessarily special or pioneering conditions. Some of these are described in the following paragraphs. Three of these, in which the furnishing of extraction steam has figured prominently, are shown in Fig. 10. The first shows one of three high-pressure, condensing, extraction, tandem-compound turbines designed to operate in power plants built and operated by the Pacific Gas and Electric Company adjacent in each case to an oil-refining plant and to furnish power and process steam to the oil refineries and excess and stand-by power to the utility transmission lines. These turbines are to operate at 1400 psi pressure, 940 F temperature, and to exhaust from zero to about 97 per cent of the full throttle flow at approximately 200 psi gage to evaporators which furnish process steam to the oil refineries. Under partial or complete condensing operation as much as 50,000 kw can be generated. This is the most extensive and efficient combined power and heat supply undertaking in which a power utility and a group of industrial plants have cooperated.

The second machine, a 15,000-kw turbine for the Iowa Electric Light and Power Company, Cedar Rapids, Iowa, is designed to supply steam to adjacent industries and power to the utilities. It is to operate at 650 psi initial steam pressure, 750 F temperature, condensing, and supply extraction steam at 100 to 225 psi gage. The machine really consists of two separate turbines in the same casing: a noncondensing turbine in the front end, and a condensing turbine on the generator end. The valve chest for one is in the upper half of the casing, and for the other in the lower half of the casing. The parallel-flow construction permits independence of operation between the noncondensing and condensing portions of the turbine and gives high efficiency at all conditions of operation.

The third machine is one of the largest noncondensing superposed turbines ever built, rated 65,000 kw, and is for the Consolidated Edison Company of New York, Inc. It will operate at 1250 psi, 925 F, and exhaust into an existing 200-psi header supplying condensing turbines and a district heating system. The low-pressure turbine between the high-pressure turbine and the generator takes a portion of the exhaust of the main turbine and exhausts it into a two-stage feedwater-heating system. The method of mounting this small turbine without bearings of its own is interesting. There will be in all four units of this general type in this plant (built by two manufacturers), two of which are operating and all of which embody the feedwater-heating turbine arrangement. This has been a nice solution to the difficulty usually associated with superposing in obtaining low-pressure steam for feedwater-heating purposes, since the existing condensing turbines of older design are often not adapted for steam extraction.

Fig. 11 shows the 3600-rpm high-pressure section and the 1800-rpm low-pressure section of a 76,500-kw cross-compound condensing and resuperheating turbine which will operate at 2300 psi, 940 F initial temperature and 900 F resuperheating temperature, in the Twin Branch station of the Indiana and Michigan Electric Company at South Bend, Ind. This company is one in the American Gas and Electric Company group and the engineering for this extension is being done by the American Gas and Electric Service Corporation. These are the highest steam conditions yet undertaken for commercial service in this country.

[11] "Preferred Standards for Steam-Turbine Generators (of 10,000 kw rating and above)." Subcommittee on Standardization of the National Defense Power Committee, Washington, D. C., Nov. 3, 1938.

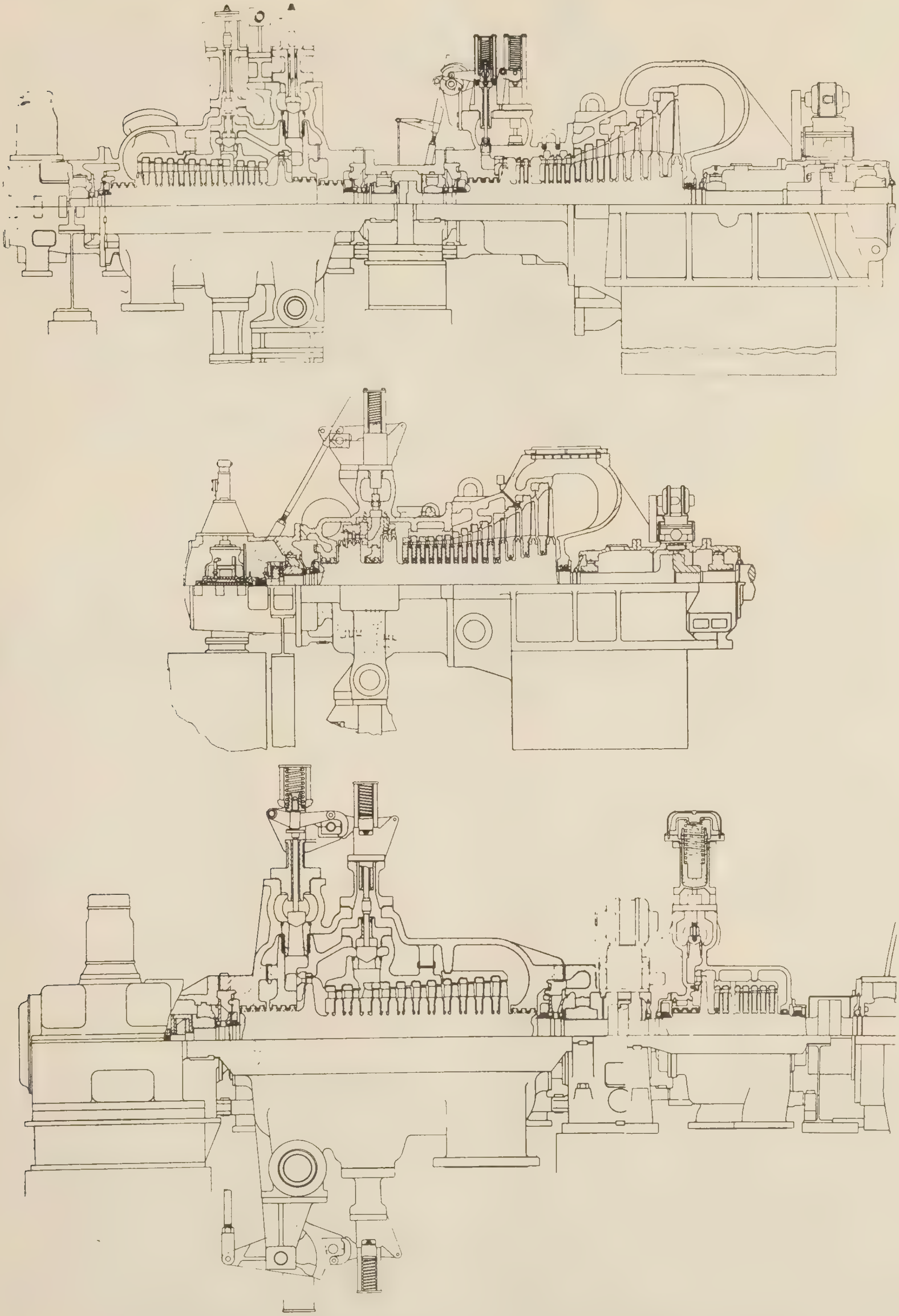

Fig. 10 Extraction-Condensing Turbines of Recent Design for High Steam Conditions

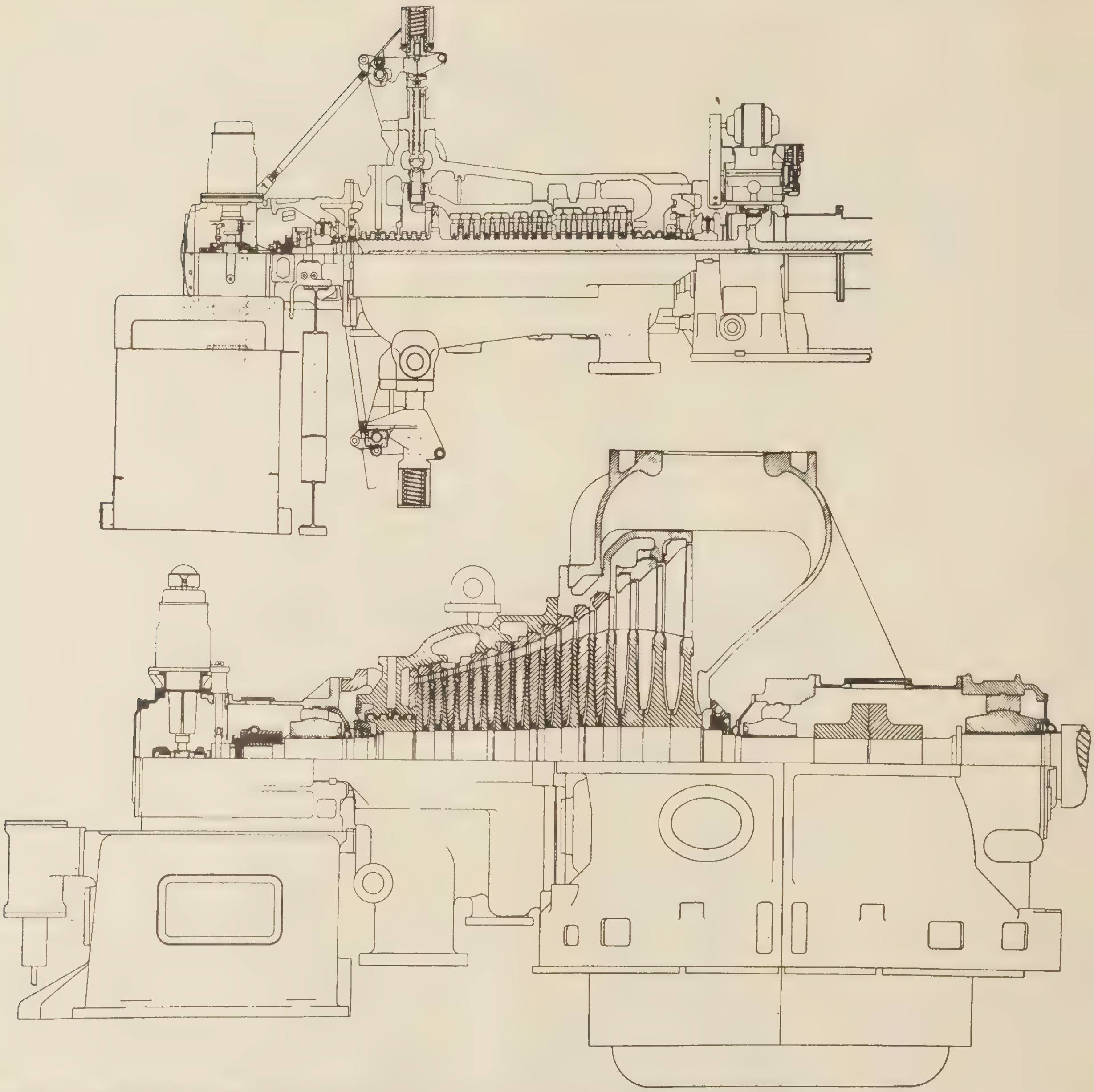

Fig. 11 Cross Sections of High- and Low-Pressure Elements of 76,500-Kw Cross-Compound Turbine for 2300 Psi, 940 F Initial and 900 F Reheat Temperature for Twin Branch Station for Indiana and Michigan Electric Company

Both of these units will drive hydrogen-cooled generators; the high-pressure turbine is of the double-shell construction and steam is extracted from the turbines at five points for feedwater heating. This installation, when in operation, should have the lowest heat consumption yet obtained from a steam power plant.

Fig. 12 shows the cross section of the high- and low-pressure sections of a 3600-rpm condensing turbine being built for the Burlington Generating Station of the Public Service Electric and Gas Company, New Jersey, which is designed for 1250 psi gage, 950 F initial conditions. These sections are each 50,000 kw, making a total rated capacity of 100,000 kw for the set, with the turbines able to carry 125,000 kw at unity power factor on the generators. In order to get adequate leaving area at 3600 rpm, it became necessary, as previously mentioned, to have four parallel flows in the low-pressure section, which discharge to a single-pass condenser arranged lengthwise under the turbine. From the condenser viewpoint, this makes an ideal arrangement in many respects, permits close spacing between the high- and low-pressure sections, and results in a very compact installation.

Normally, these two turbines operate as a single cross-compound turbine with the cross-over pressure fluctuating with load. The sections are valved in such a way as to permit emergency operation of either one separately. The high-pressure section may discharge into a steam header at 200 psi pressure. The low-pressure section may operate either through a reducing valve from the high-pressure boiler, or from the 200-psi steam header. Both generators will be hydrogen-cooled.

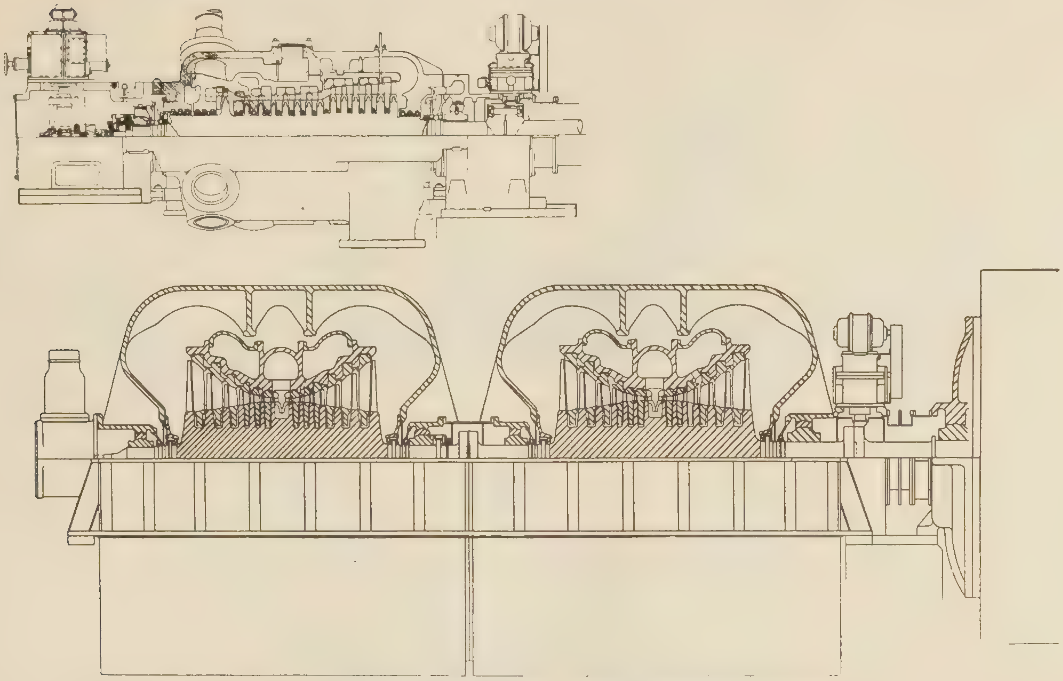

Fig. 12 Cross Sections of High and Low Elements of 100,000-Kw Cross-Compound 3600-Rpm Condensing Turbine for 1250 Psi and 950 F Initial Temperature for Burlington Generating Station of Public Service Electric and Gas Company, New Jersey

Fig. 13 shows a cross section of the third of three similar vertical compound turbines in operation in the power plant of the Ford Motor Company at River Rouge, Detroit. This unit recently put in service has hydrogen-cooled generators. The first of these turbines operates at 750 F with steam resuperheating between the two units; and the second and third operate at 1250 psi, 900 F initial temperature, with resuperheating between the units eliminated. Test results are shown on the last two of these units in a later portion of this paper. The first two of these vertical compound machines have had a very remarkable record from the standpoint of reliability. The third, of course, has not yet been in operation long enough for its record to have any significance.

Records submitted by the Ford Motor Company complete up to March, 1940, are shown in Table 1.

Fig. 14 shows the condensing turbine, previously mentioned, now being designed for 25,000 kw normal capacity, condensing, 1300 psi initial pressure, to operate initially at 960 F, utimately at 1000 F temperature. It is the first single-cylinder, double-shell, condensing turbine which has been designed. The highest temperature to be encountered by the rotor, by the turbine diaphragms, or by the main bolt of the turbine shell and the horizontal joint bolting will be about 890 F at 1000 F initial temperature and maximum load. This turbine will be installed in the Atlantic City Electric Company's plant, which is in the South Jersey System of the American Gas and Electric Company.

TABLE 1 RECORDS OF FORD MOTOR COMPANY UNITS

Units Nos. 1 and 2	
(First vertical compound unit, officially started July 23, 1931, by Mr. Henry Ford)	
Total hours installed	75,412
Total hours in operation	66,044
Total hours down due to no load	9,032
Outage due to reheater repair, hr	336
Total generated kwhr	2,932,965,400
Availability, per cent	99.5
Units Nos. 3 and 4	
(Second vertical compound unit, officially started July 15, 1936, by Mr. Henry Ford)	
Total hours installed	31,728
Ootal hours in operation	28,368
Total hours down, due to no load	3,360
Tutage	None
Total generated kwhr	1,002,663,000
Availability, per cent	100

Rebuilding of Old Machines

A number of utility companies have found it profitable to undertake rather extensive rebuilding of old machines. Some of the rebuildings were for greater capacity and higher efficiency at the same steam conditions; some to change the frequency from 25 cycles to 60 cycles, and at the same time secure additional capacity; and some to obtain additional capacity, higher efficiency, and to operate at higher steam conditions.[12] These types of rehabilitation are of importance particularly in times of depressed business conditions when few new major plants are under consideration.

Sometimes older turbines are rebuilt or modernized to give them a new "lease on life," particularly when they are intended to be used from then on as the low-pressure elements of a new superposed turbine.[13]

This rejuvenating of old power equipment has undoubtedly extended the useful life, at relatively high load factors, of much equipment which was formerly considered obsolete and good only for peak-load service.

One interesting rebuilding proposition undertaken and completed was the furnishing of complete new internal turbine elements for a 160,000-kw cross-compound European-built turbine in the Hell Gate plant of the Consolidated Edison Company of New York, Inc.

[12] "Modernizing the Connors Creek Power Plant," by S. Crocker, *Combustion*, vols. 6 and 7, 1935, p. 11.

"Load-Area Detroit Plant," by P. W. Thompson, *Electrical World*, vol. 106, 1936, p. 2685.

[13] "Logan Steam Plant Landmark," by Philip Sporn, *Electrical World*, vol. 106, 1936, p. 1017.

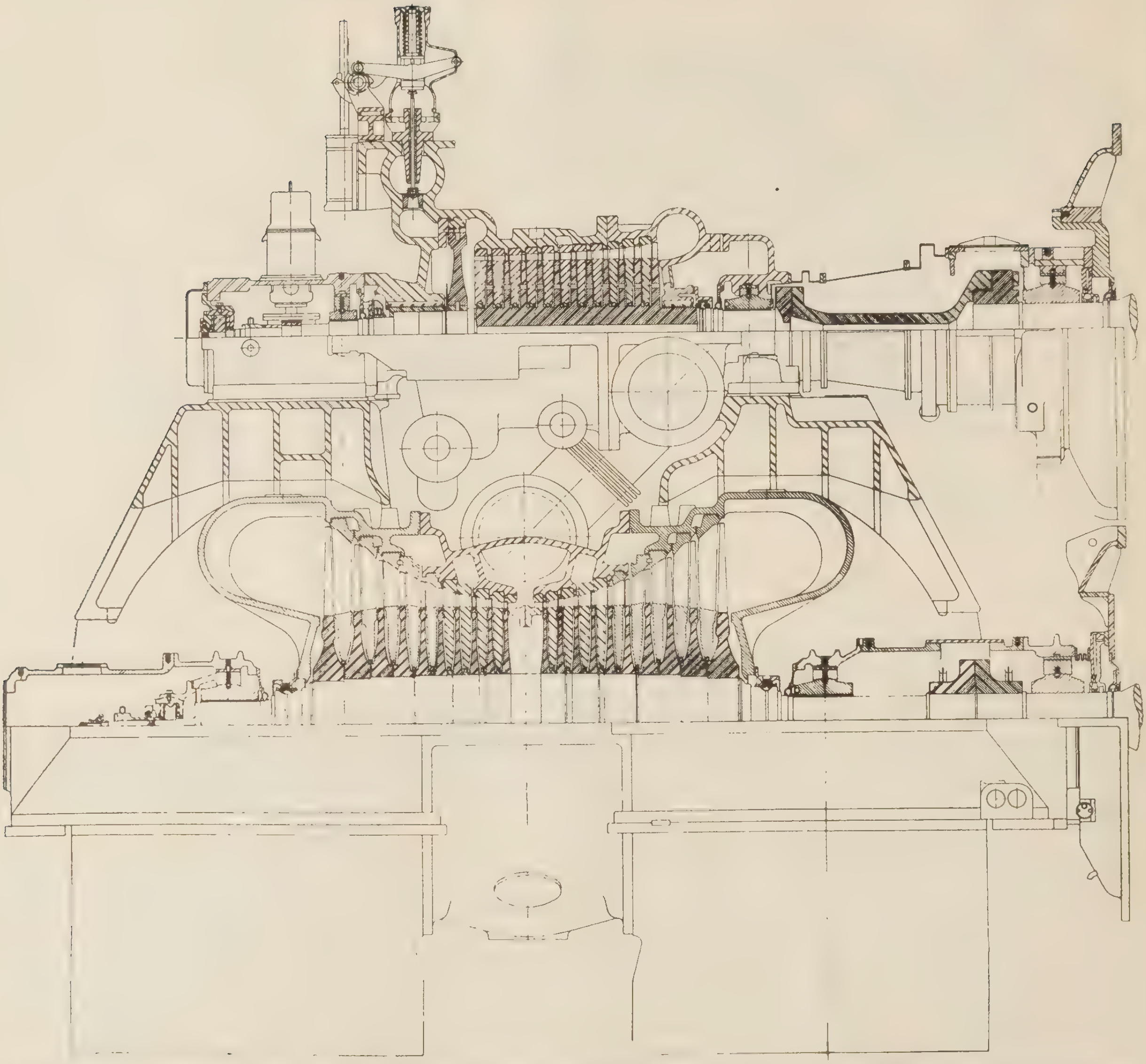

Fig. 13 Cross Section of Third 110,000-Kw Vertical Compound Turbine Installed in the River Rouge Power Plant of the Ford Motor Company

Fig. 15 shows cross sections of the rebuilt high- and low-pressure elements of this machine as compared to the original design.

Fig. 16 shows the recent test results of the rebuilt machine as compared with tests of the original machine. The generators are not of modern design and efficiency, and the turbines as rebuilt had a comparatively small number of stages and shorter last-stage buckets than the original.

The Trend of Reliability

The trend of turbine reliability has been toward improvement despite the higher steam conditions and the higher operating speeds. The figures on turbine outage compiled by the Edison Electric Institute each year and published in the Annual Reports of the Turbine Subcommittee of the Prime Movers Committee[14] support this view, as shown in Fig. 17.

[14] "Turbines, Condensers, and Pumps 1939 (A Report of the Turbines Subcommittee of the Prime Movers Committee, Edison Electric Institute)." Publication No. G7, published Jan., 1940.

Certain difficulties have been experienced on some recently constructed turbines, largely in the first stage of high-capacity 3600-rpm superposed turbines, which have increased the outage of these machines during the last year or so. It is felt, however, that these difficulties are now fairly well solved, and that this factor should not be a major consideration on these turbines or on new turbines in the future.

In spite of this difficulty, however, on eight turbines of this type now in operation ranging in capacity from 10,000 kw to 60,000 kw with a total of 17 machine-years, up to Jan. 1, 1940, the unscheduled outage due to all turbine difficulties has been 6.7 per cent. The unscheduled outage in the last nine months of 1939 has been 3.6 per cent, which is less than the average of all condensing machines.

Bucket troubles or bucket failures used to be a greater factor in turbine outage. Figures published in the 1939 report of the turbine subcommittee already referred to indicate that the outages from this cause on all turbines reported have been

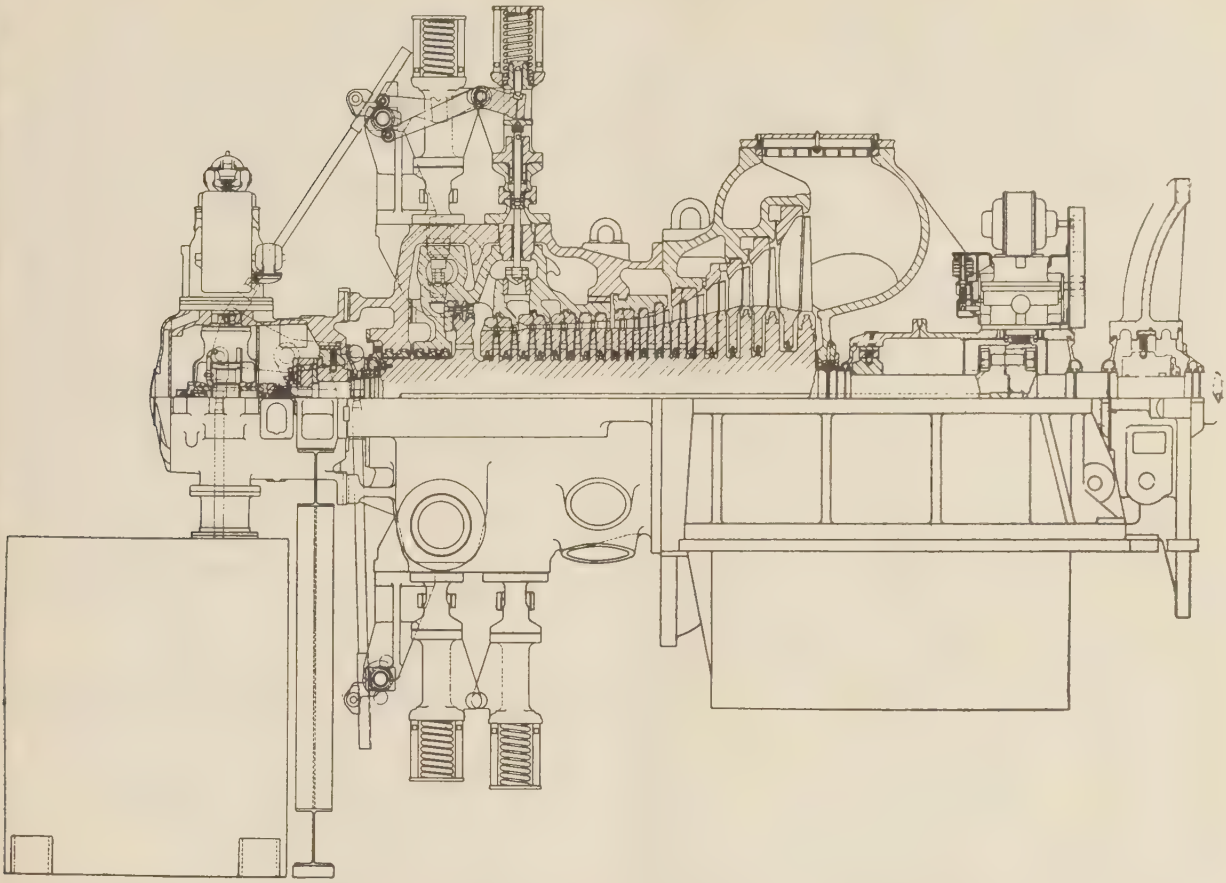

Fig. 14 25,000-Kw, 3600-Rpm Condensing Double-Shell Turbine Designed for 1350 Psi, 1000 F Initial Temperature, to Be Installed by the Atlantic City Electric Company, American Gas and Electric Company

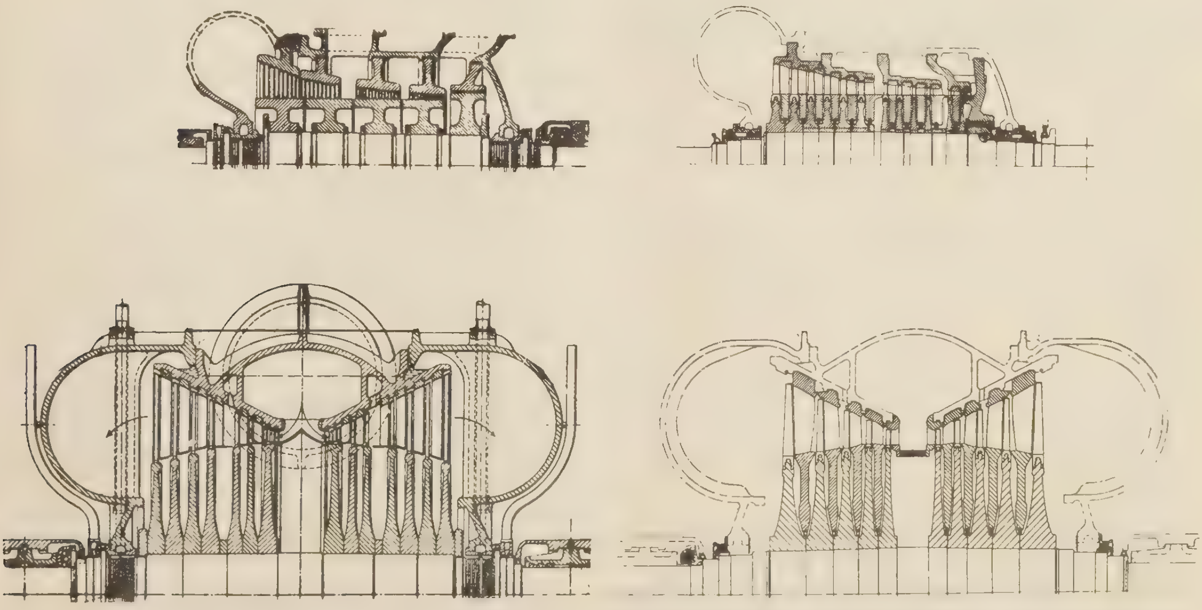

Fig. 15 Cross Sections of High- and Low-Pressure Elements of Original and Rebuilt 160,000-Kw Turbine at the Hell Gate Station of the Consolidated Edison Company of New York, Inc.

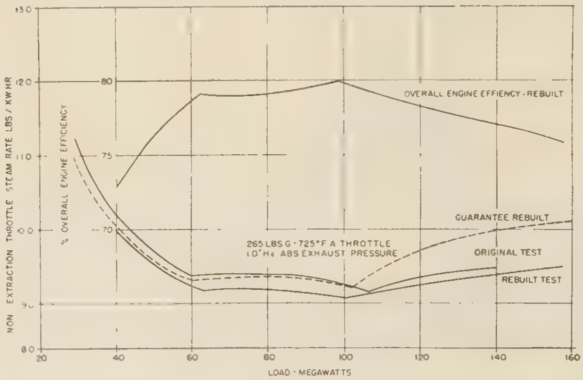

Fig. 16 Test Results of 160,000-Kw Turbine Shown in Fig. 15

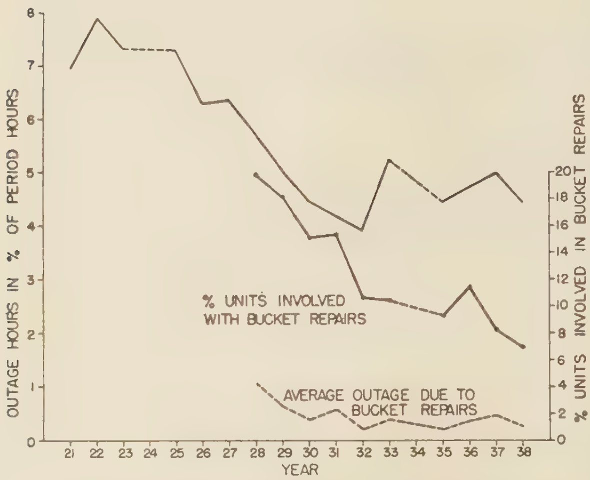

Fig. 17 Analysis of Turbine Outage Including Normal Inspections on All Makes and on All Turbines Reported to Turbine Subcommittee of the Prime Movers Committee, Edison Electric Institute, Based on 1939 Report

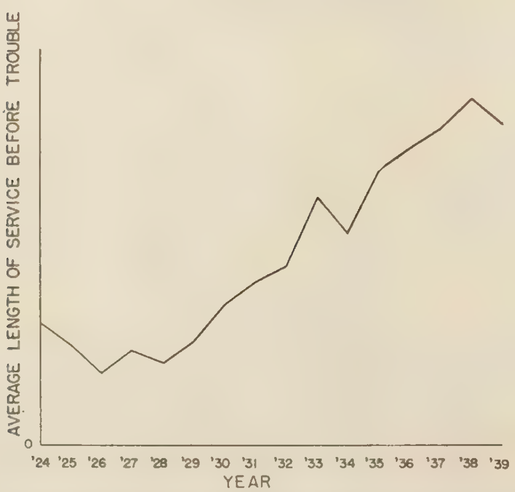

Fig. 18 Average Length of Service of Original Rows of Turbine Buckets on Which Trouble Was Reported

reduced to less than half what they were ten years ago. This is also shown in Fig. 17.

Fig. 18 is a curve covering the last 16 years which shows the average operating life of the original bucket rows in turbines built by the author's company in each year on which difficulty was experienced. This is the average operating life of those buckets only on which trouble occurs. It will be noted that with the exception of this last year, in which the influence of the first-stage bucket troubles previously mentioned have been a contributing factor, and one other year, this curve has gone up steadily every year since 1924, and the average life of the buckets which gave trouble has been tripled before trouble is encountered.

One additional difficulty to which the newer larger 3600-rpm turbines have been subject during the last two years has been in connection with thrust-bearing failures or outages ordered for the purpose of permitting changes to prevent future thrust failures. Among other things contributing to this difficulty was the fact that the greater "out-tangent" effect of the nozzle jet on the smaller-diameter wheels lowered the pressure on the entrance side of the wheel below the values which had previously been calculated. This produced more thrust of the turbine rotor against steam flow than had been provided for. Tests and analyses made since have, it is believed, permitted a much better evaluation of this situation and should prevent future trouble.

The Trend of Turbine Economy

The economy with which power can be produced by steam from fuel has steadily increased. This has been a result of three major factors:

1 Improvement in steam conditions.

2 Improvement of the heat cycle through which the steam is used.

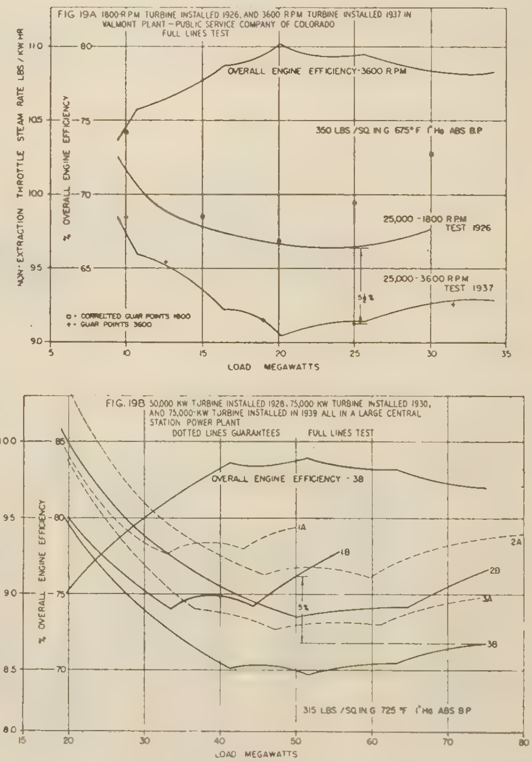

Fig. 19 Comparative Steam Consumption of Various Turbines

3 Improvement in the basic "engine efficiency ratio" itself, that is, improvement in the percentage of the available energy in the steam which the turbine makes usable.

The improvement in the turbine or engine efficiency ratio has resulted from a continuous and comprehensive program of research and design studies upon the various elements by means of which the energy in the steam is turned into useful energy in the turbine. Losses at every point have been studied and carefully reduced.

It is, of course, somewhat difficult in the over-all results to sort out the separate influences of the foregoing factors, but some indication will be given in the following comparison of several turbine tests and guarantees made over the last few years on turbines built by the author's company.

Figs. 19*A* and 19*B* show the results of tests on successive turbines in the same plant operating at the same steam conditions.

Fig. 19*A* shows the nonextraction tests and the corresponding guarantees on two turbine generating sets installed in the Valmont plant of the Public Service Company of Colorado; one a 25,000-kw unit installed in 1926, and the other a 25,000-kw 3600-rpm unit installed in 1937. Both operate at the same steam conditions and have air-cooled generators. The reduction in steam consumption when operating nonextraction is about 5 per cent to 6 per cent, and the total reduction in heat consumption due to the more efficient feed-heating cycle on the second turbine would be even greater.

Fig. 19*B* shows the nonextraction steam rates of three turbines installed in a large central-station power plant; the first a 50,000-kw turbine installed in 1927, the second a 75,000-kw turbine installed in 1929, and the third a 75,000-kw turbine installed in 1937. These turbines are all 1800-rpm machines, equipped with air-cooled generators, and operate at the same steam conditions. A 5 per cent gain was made between the first and the last machine installed.

Fig. 20 shows the comparative results of successive turbines in three different stations.

Fig. 20*A* shows a comparison between the heat-consumption guarantee on the first 110,000-kw vertical-compound turbine generator installed in the River Rouge plant of the Ford Motor Company in 1931, on which a complete test was not made, and the guarantees and tests of the second and third machines installed in this plant in 1936 and 1939, respectively. All are of the same capacity and operate at the same steam pressure. The first machine had steam-heated resuperheaters, but on the second

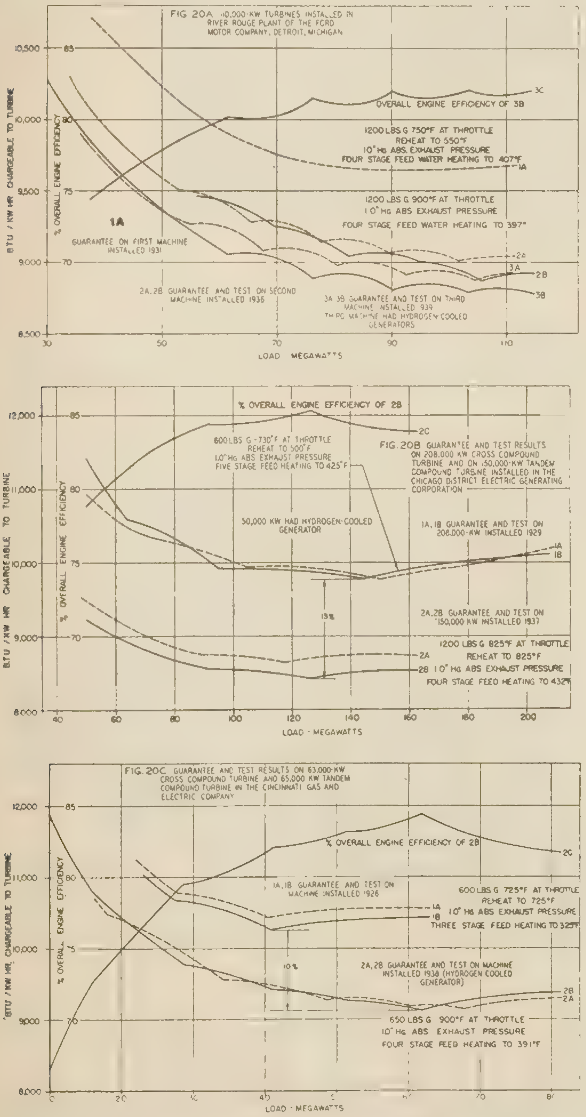

Fig. 20 Comparative Heat Consumptions of Various Turbines

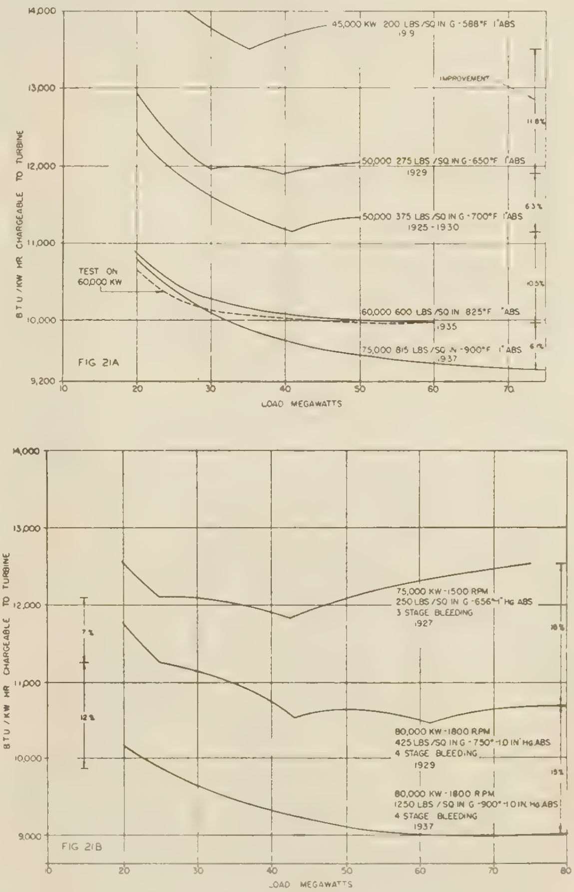

Fig. 21 Equivalent Comparative-Heat Consumption Guarantees on Successive Turbine Generators Sold to Two Different Public-Utility Companies

and third machines the initial temperature was raised so high resuperheating could be dispensed with.

Fig. 20*B* shows the guarantees and test results of a 208,000-kw cross-compound turbine installed by the Chicago District Electric Generating Corp. in 1929 operating at 600 psi, 730 F, resuperheat temperature 500 F; and the guarantee and test results on the recently installed tandem-compound 150,000-kw turbine operating at 1200 psi, 825 F, resuperheat temperature 825 F, driving a hydrogen-cooled generator and installed in the same station.

This latter turbine generator probably has the lowest heat consumption so far obtained, consuming 8460 Btu per kwhr at the most economical load. The reduction in heat consumption between the 208,000-kw and the 150,000-kw set is about 13 per cent.

Fig. 20*C* shows the test results of a 63,000-kw cross-compound 1800-rpm turbine-generator unit operating at 600 psi, 725 F initial temperature and 725 F reheating temperature, installed in 1926 in the Columbia Park station of the Cincinnati Gas and Electric Company, and rebuilt in 1928; as compared with the test results on a recently installed 65,000-kw tandem-compound turbine equipped with a hydrogen-cooled generator, operating at 650 psi, 900 F initial temperature without reheat, installed in 1938. The improvement is about 10 per cent. This does not reflect against the value of resuperheat, but shows the advantage of higher temperature, better turbine design, lower leaving loss, and the gain from hydrogen cooling. If resuperheating were used the economy would be even better.

Fig. 21 shows the heat-consumption guarantees on successive turbine generators sold to two different public-utility companies over the last 25 and 13 years, respectively. The differences between the first and the last guarantees are some 35 per cent and 27 per cent, respectively.

Performance of Noncondensing Topping Units

Tests have been run recently on six large high-pressure, high-temperature, noncondensing turbines. With one exception, these turbines have, in general, met their contract guarantees. The inherent difficulties of making such tests, and of insuring that the turbines are free from boiler solids in new plants have not yet permitted obtaining the accurate and uniform results which are possible with condensing machines.

However, over-all test efficiencies of noncondensing turbines operating between 1200 psi and 200 psi pressure, based on the net energy output of the generator, and the steam conditions ahead of the stop valve, were about 79 per cent on 10,000-kw sizes with air-cooled generators, and about 84 per cent on 50,000-kw turbines with hydrogen-cooled generators.

These engine efficiency ratios are generally greater for the range of energy through which these turbines are operating than the over-all engine efficiency ratios of the low-pressure turbines into which they are discharging.

These results, together with those shown in Figs. 19, 20, and 21, showing results on both high-pressure and low-pressure turbines, indicate quite definitely that the expected performances of the newer high-pressure turbines are being realized.

Cleanliness of Internal Parts of Turbines

The importance of the maintenance of internal cleanliness on turbines during operation cannot be overemphasized. It is of no value for the turbine designer to refine the nozzle and bucket shape and then to have solids deposit out from the steam and reduce the high efficiency so obtained. It is an action similar to the more well-known icing up of airplane wings and propellers.

Variation in load and periodic shutdowns tend to wash the deposits out; whereas, base-load machines are apt to build up deposits.[15] Washing procedures are available which, if carefully carried out, will restore the machine to reasonable cleanliness if the deposits are soluble.

The problem can usually be solved at the source as evidenced by the large number of stations in which the turbines seem to operate for years without difficulty. In other stations deposits build up so rapidly that it is impossible to obtain a reliable test immediately after an inspection.

The solution seems to lie in proper treatment of the feedwater and control of carry-over from the boiler drums under each local condition encountered. New stations are likely to have this difficulty until these problems are solved for their particular conditions.

PART III—DETAIL CONSIDERATIONS OF TURBINE DESIGN

Number of Stages and Cylinders or Casings

The proper number of stages for best economy and the proper number best to meet the other requirements of design and first cost have been a question on which turbine designers have differed greatly among themselves and from time to time. It is now quite definitely established, on the basis of both research work and analysis from fundamental hydrodynamic principles, that turbine stage efficiencies generally:

(*a*) Increase as the radial height of the nozzle and bucket of the stage is increased, other conditions remaining fixed.

(*b*) Are relatively constant for a given stage with varying steam velocity (if the ratio of bucket to steam speed is held constant), but tend to be somewhat lower at lower velocities and highest near the velocity of sound.

(*c*) Are likely to decrease with smaller wheel diameters, bucket length and speed remaining constant.

Specific nozzle and bucket designs may change these relations, but they are generally safe rules to follow with contemporary design practice. Item (*a*) indicates that if the number of stages in a turbine is increased, and, hence, since the total bucket speed is limited, the diameters are reduced, and the bucket lengths increased, then the efficiency will be increased. On the other hand, (*b*) and (*c*) indicate that some of this gain may be lost by the lower steam velocities and smaller wheel diameters so encountered.

Furthermore, in an actual turbine design, experience seems to indicate that the diaphragm and high-pressure packing radial clearances which can be maintained between the rotating and stationary elements are a function of the span between bearings, actually appearing to be, based on measurements on modern turbines equipped with turning gears and after good commercial operation, about *2 mils per foot of shaft span* between bearing center lines. This factor puts a rather definite limit at around 20 to the number of stages which can be carried advantageously between one pair of bearings.

Where a turbine must be double flow in the low-pressure turbine, it can be divided advantageously between two different bearing spans; but to divide a single-flow machine in order to secure the increased efficiency of the larger number of stages and the shorter, smaller shafts so permitted, may introduce other losses, such as cross-over pressure drop, and extra bearing and external packing losses, so as to neutralize much of the gain except in turbines for higher pressures.

The result of all of these factors has been that it has been possible to develop compact single-cylinder turbines which compare very favorably with two- and three-cylinder tandem-compound turbines both as to economy and reliability.

[15] "Power Station Chemistry 1937 (A Report of the Power Station Chemistry Subcommittee of the Prime Movers Committee, Edison Electric Institute)." Publication No. E10, September, 1937.

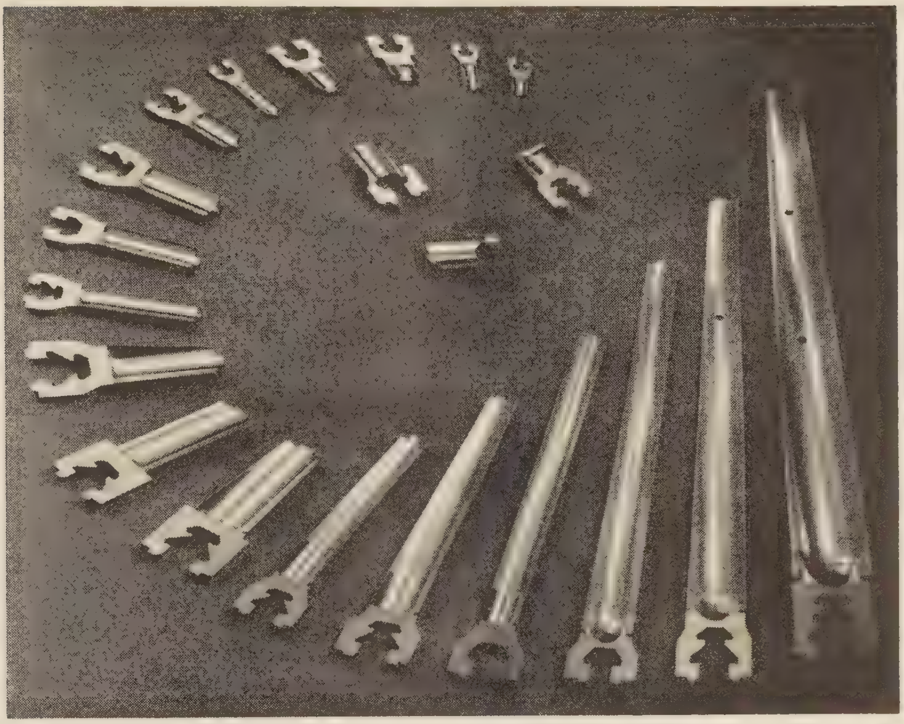

Fig. 23 Typical Group of Buckets

Nozzles and Diaphragms

Nozzles and diaphragms are the transverse walls which separate the stages, withstand the stage pressure drop, and carry the directing nozzles through which the pressure drop is changed to

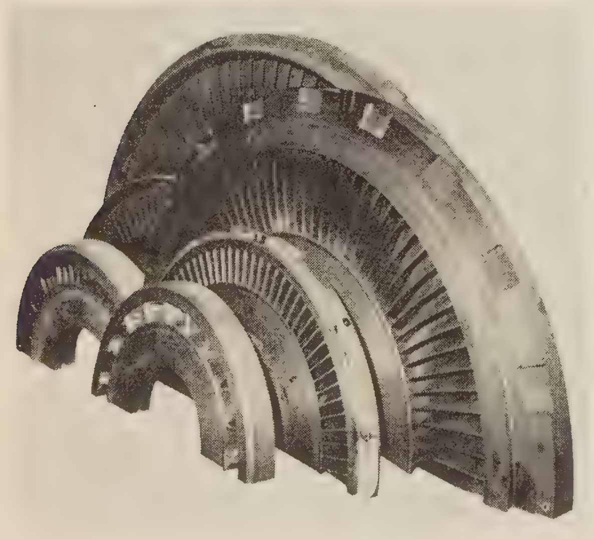

Fig. 22 Typical High- and Low-Pressure Diaphragms

a speed increase in the steam stream, which is in turn directed onto the moving buckets by the nozzles.

These elements are all important from the standpoint of turbine efficiency, and have been the object of a comprehensive research program both in this country and abroad extending over the last twenty years, and still being actively pursued.[16] This research work has resulted in a great improvement in the efficiencies of the nozzles used in actual turbines and hence in an improvement in the over-all efficiency of the turbine generator set.

The mechanical design of these elements is also of equal and coordinate importance. They must be strong enough to withstand terrific mechanical forces of both a steady and a highly periodic character, ofttimes at high temperatures; and, since they carry the diaphragm packings, they must stay accurately centered with the shaft. In order to do this effectively they

[16] "The Turbine Designer's Wind Tunnel," by H. L. Wirt, *Mechanical Engineering*, vol. 47, 1925, pp. 13–17.

"A Machine for Testing Steam-Turbine Nozzles by the Reaction Method," by G. B. Warren and J. H. Keenan, *Mechanical Engineering*, vol. 48, 1926, pp. 227–232.

The six reports of the Steam-Nozzles Research Committee:

- First report, Proceedings of The Institution of Mechanical Engineers, 1923, vol. 1, January-June, p. 1.
- Second report, Proceedings of The Institution of Mechanical Engineers, 1923, vol. 1, January-June, p. 311.
- Third report, Proceedings of The Institution of Mechanical Engineers, 1924, vol. 1, January-May, p. 455.
- Fourth report, Proceedings of The Institution of Mechanical Engineers, 1925, vol. 2, May-December, p. 747.
- Fifth report, Proceedings of The Institution of Mechanical Engineers, 1928, vol. 1, January-May, p. 31.
- Sixth report, Proceedings of The Institution of Mechanical Engineers, 1930, vol. 1, January-May, p. 215.

"Some Researches on Steam-Turbine Nozzles Efficiency," by H. L. Guy, presented at the Institution of Civil Engineers, London, 1939, Sir Charles Parsons Memorial Lecture.

"An Investigation of Energy Losses in Steam-Turbine Elements by Impact Traverse Static Test With Air at Subacoustic Velocities," by Winston R. New, Trans. A.S.M.E. vol. 62, 1940, pp. 489–502.

"Automatic Integrating Pressure Traverse Recorder for Study of Flow Phenomena in Steam-Turbine Nozzles and Buckets," by H. Kraft and T. M. Berry, Trans. A.S.M.E., vol. 62, 1940, pp. 479–488.

must be supported so as to stay centered within 10 mils, if possible, despite total heat expansions at their outer diameters of 30 times this value. Fig. 22 shows typical nozzle diaphragms for high- and low-pressure sections of the turbines.

The manufacture of these parts is a highly specialized process. It differs radically for the higher- and lower-pressure diaphragms. The former are fabricated by welding and the latter by casting.

First-stage nozzles are subjected to particularly difficult conditions because of: (1) High pressure drop, particularly at lighter loads; (2) the high temperature at which they operate; and (3) the requirements of steam flow which make it undesirable to increase the exit edge thickness unduly. Furthermore, it has been found that the first-stage buckets react on the nozzle edges, and so produce violent alterations of the pressure drop across the nozzle edges, and at frequencies which may reach from 6000 to 20,000 per second.

Buckets

Buckets are the elements of turbines which are attached to the wheels or rotor and which convert the energy in the moving steam stream into power on the wheel rim.

The more general problems of last-stage bucket designs have been discussed in Part II of this paper because of the immediate bearing upon the entire turbine design.

In the shorter impulse buckets, the basic design from the steam standpoint has changed but little in 35 years despite the results of much research work. This is probably due less to the present excellence of the general design than to the inherent difficulty of carrying out the research and isolating the various factors, since the moving buckets and stationary nozzles are so interrelated. Research now in progress may alter this situation, however.

Longer buckets, owing to inherent lower steam leakage, are generally improved in efficiency by being designed for some pressure drop across them, partaking, particularly at the tip, of nozzle characteristics, and so have been improved by the years of highly fruitful nozzle research.

With buckets the problems of mechanical design and manufacture have been of paramount importance, particularly from the standpoint of turbine reliability. Here perfection has been obtained only by painstaking research[17] into the vibration characteristics of such structures, supplemented by the use of design strength factors based upon a complete statistical record and analysis of past bucket troubles and past successful operation.

Fig. 23 shows a group of typical buckets for both 1800-rpm and 3600-rpm turbines. The particularly rugged characteristics of the design and attachments of some of the buckets should be noted, as well as the difference in character between the root and tip shapes of the longer buckets.

The shape and proportions of the attachments or "dovetails" have been worked out analytically, supplemented by photoelastic-stress analysis and pull and vibration tests to destruction of full-size samples.

Finally, all large 1800-rpm bucketed turbine wheels and typical 3600-rpm wheels are tested for vibration characteristics and tuned to run "off resonance" in the wheel-testing machine shown in Fig. 24. More than 110,000 oscillograph films similar to that shown have been obtained, analyzed, and recorded.

Buckets which cannot be protected by this tuning method, which in general includes most buckets under 10 in. in length, must be protected against vibration by being made strong enough to run, if necessary, "on resonance."

[17] "The Protection of Steam-Turbine Disk Wheels From Axial Vibration," by Wilfred Campbell, Trans. A.S.M.E., vol. 46, 1924, pp. 31–60.

"Tangential Vibration of Steam-Turbine Buckets," by Wilfred Campbell and W. C. Heckman, Trans. A.S.M.E., vol. 47, 1925, pp. 643–671.

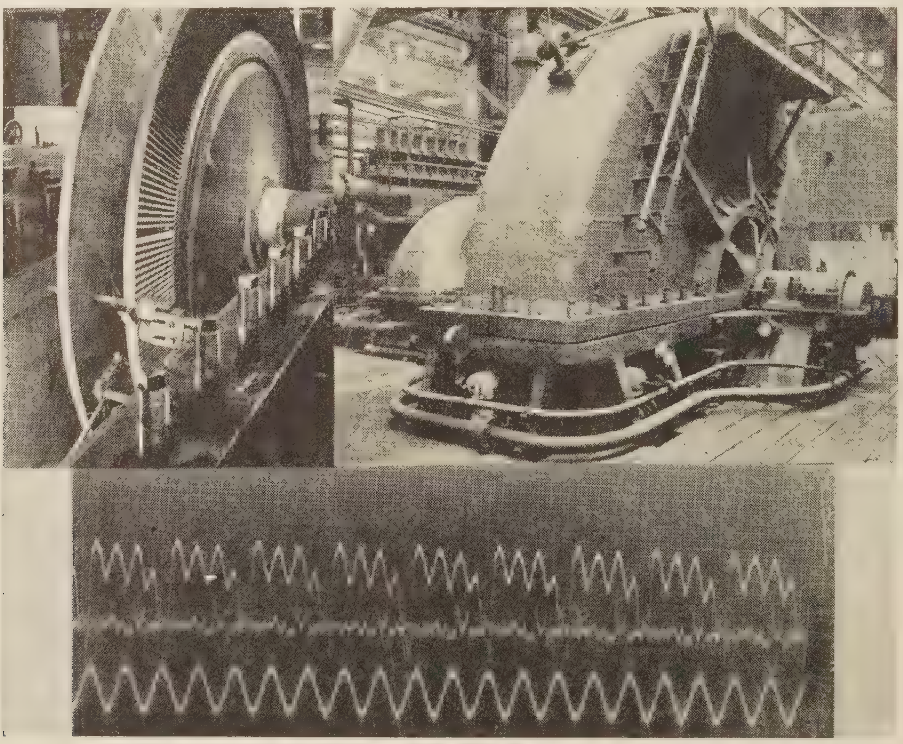

Fig. 24 Wheel Vibration Testing Machine With Typical Oscillograph Record of Tests

Tests indicate that for a given bucket wheel the amplitude of resonant vibration generally increases in proportion to the load being carried. Statistical experience accumulated by an analysis of results obtained on about 10,000 wheel-years of operation indicates quite definitely that the liability of reaching vibration stresses in the buckets on such wheels which will produce failure is reduced to the vanishing point when the buckets are made strong enough in relation to the loads carried. It is essential, of course, that the other details of design and construction be uniformly and properly carried out.

Rotors

The rotor, aside from the buckets, might be considered as the aggregate of the shaft and the wheels which carry the buckets, together with the rotating elements of the packings.

Rotors for 1800-rpm turbines are generally made of a shaft with wheels separately shrunk on and packing rings. The wheels in the low-temperature section are keyed to the shaft. In the higher-temperature sections it has been found necessary to be sure that the wheels are able to leave the shaft under the combined influence of centrifugal force and a sudden increase in temperature, and at the same time stay central and drive the shaft. This is accomplished by a so-called "pin bushing." The bushing is keyed to the shaft, and the wheel hub is attached to the bushing itself by a number of radial pins.

Two refinements which have resulted in great improvement in the operation of this type of turbine are: (1) Making the keys discontinuous, that is, a separate key for each wheel, which does not extend under the packing ring between adjacent wheels; and (2) the undercutting of the packing sleeves to reduce the heat passed into the shaft due to accidental rubbing of the packings.

3600-Rpm Turbines

Rotors for 3600-rpm turbines are generally machined from a solid forging. This obviates the difficulties associated with separate wheels and packing sleeves, and secures a short rigid rotor which permits operation of such high-speed turbines below the calculated critical speed. These rotors have shown great stability under changing load and temperature conditions.

This rotor construction also possesses an important inherent advantage in the ease with which it is heated due to its extended surface. This gives it an automatic "end loosening" effect in that during starting the axial clearanes are opened up without change of the thrust-bearing position. The close proximity of the diaphragms prevents the rotor's cooling faster than the shell on shutting down and so prevents a tightening of the clearances.

A refinement in manufacture which has materially contributed to smoother operation of turbines under service conditions is the so-called "heat stabilizing" of the shafts. Every shaft or rotor on the turbines under consideration receives this test and treatment which is as follows:

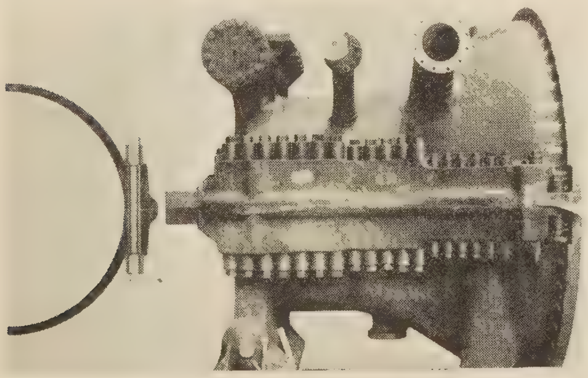

Fig. 25 Typical Horizontal Joint Flange and Bolting

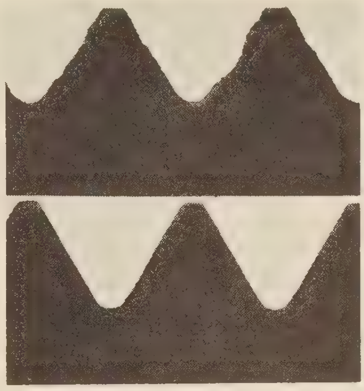

Fig. 26 Magnified Photograph of Turned and Milled Threads on Stud

After a shaft or rotor is rough-machined it is put in a special lathe in an electrically heated furnace and slowly heated while being revolved, and it is carefully indicated for "runout" during the heating. Most normal shafts will increase their runout 5 to 15 mils in this process until a temperature of 700 F to 900 F is reached, at which some kind of a readjustment takes place, probably on the machined surfaces, which permits the shaft to come back to a "hot runout" of not more than 1 to 2 mils more than its "cold runout," and on being cooled its runout does not change greatly. On putting such a shaft through a second heating its runout will not go through this cycle, but will go to the hot-runout condition directly. Some shafts will have a large hot runout which cannot be corrected this way. Such a shaft or rotor must be reheat-treated, and if this does not correct the condition, it must be rejected, otherwise the balance of the turbine in which it is used would be seriously different in the hot and cold condition and would change with load.

All wheel and solid rotor forgings must be metallurgically sound, and are checked following final machining by the magnaflux method.

Bolting Flanges

The horizontal flanges by which the upper and lower halves of the shells are joined are perhaps one of the most important parts of the turbine design. This joint must remain tight despite temperature and pressure conditions and changes. It must permit of being dismantled, and reassembled, and still remain tight; and this without the aid of gaskets.

Fig. 25 shows a typical shell flange. The bolts are put in as close to the inner diameter as possible, and still leave sufficient metal between the inner flange wall and the bolt hole, generally around 1 in. as a minimum. This requires a flange of great depth. The flange is given a sizeable "toe" or portion beyond the bolt. Studies carried out several years ago on rubber models indicated definitely that this configuration resulted in relatively low bolt stresses while at the same time it kept the joint tight at the cylinder bore.

The nuts are made cylindrical, with the "hex" on top, which permits a smaller hex and hence smaller wrenches. This in turn permits a closer spacing of the bolts than is possible with nuts of the conventional type.

Joints of this type when properly "scraped to fit" appear to hold steamtight when the stress in the bolts is about $1^1/_2$ to 2 times that required to overcome the bursting pressure on the

inner cylinder diameter. The "softness" or elasticity of the flange resulting from its depth and high compressive loading must make the two faces conform to each other much as two pieces of rubber would at pressures within our ordinary experience.

In the design of high-pressure turbine shells it is important

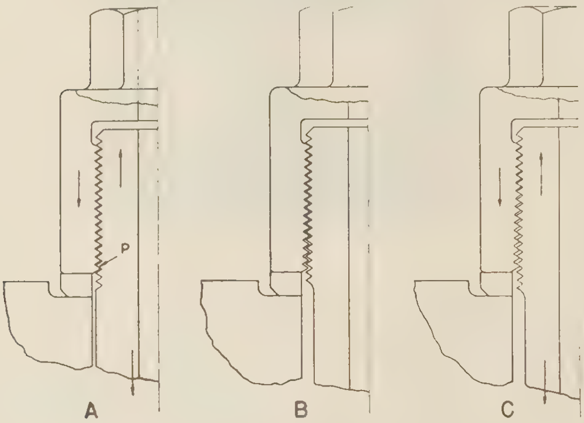

FIG. 27 TAPERED THREADS TO DISTRIBUTE LOAD ON THREADS OF HEAVILY LOADED STUDS

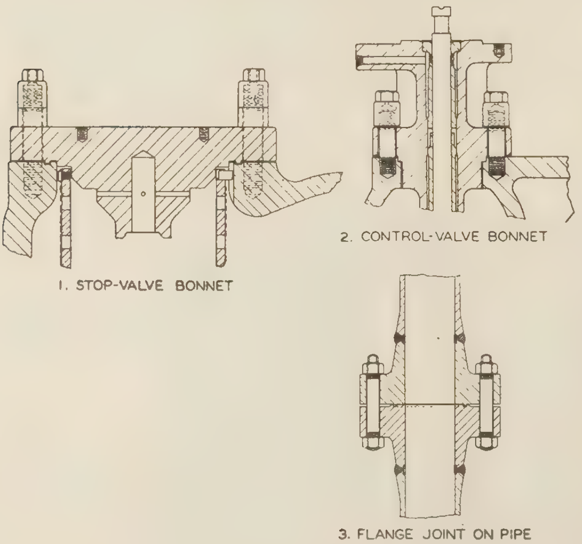

FIG. 28 CIRCULAR FLANGE JOINTS FOR HIGH PRESSURES AND TEMPERATURES

that abrupt changes of diameter be avoided as much as possible, particularly to avoid a small "waist" between two larger sections because the normal permanent distortions which accompany heating and cooling are of such a nature as to cause the flange to want to open up at this small waist portion. Also, projecting inner walls or rings attached to the shells should be avoided if possible because they tend to heat up faster than the shell and so by expansion, force the joint open.

A manufacturing refinement which has been introduced into the manufacture of the bolts and studs is to mill these threads on a special machine rather than to turn them as was formerly done. Fig. 26 shows a magnified section of the threads on studs on which the threads have been turned, and on which they have been milled. The superior quality of the milled thread is apparent.

Two further refinements are: (1) To plate the threads of the nut with a thin copper coating; and (2) to make the clearance between the nut and stud somewhat larger than normal, about 5 mils on 1-in-diameter and 10 mils on 3-in-diameter studs. Both refinements seem conducive to ease in removing nuts from studs after use.

All studs over 2 in. in diameter are made hollow to permit heating for ease in setting up and loosening.

A refinement recently introduced is illustrated in Fig. 27. This shows, first, a nut and stud combination in which the threads are of the usual parallel construction, and second, a nut and stud in which the threads on the nut are machined on a taper, shown exaggerated in the figure, of course. In the nut with the parallel threads the load must of necessity be concentrated on the first few threads, if not even on the first, because of the tendency of the remaining portion of the stud to stretch away from, and the nut to compress away from, the load with resultant overstressing of the portion of the stud adjacent to this first thread. The tendency of the nut to stretch circumferentially tends to reduce this concentration,[18] but the freedom to stretch circumferentially is reduced in the nuts on studs which are set up by heating. In studs screwed into castings rather than into nuts, such concentration of loading is very pronounced. This concentration of stress has caused some studs to crack at the bottom of the nut or where entering the casting. If, however, threads on the nut or casting are cut on a proper taper, as shown, the threads at the end of the stud are loaded first, and when tight the loading is more uniformly distributed. The taper is sometimes put on the threads of the stud. Such tapered threads also reduce the concentration of stress if the studs are not perfectly aligned with the threaded holes.

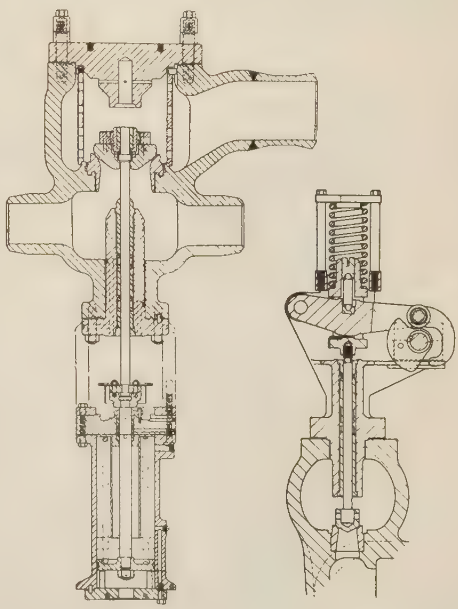

FIG. 29 TYPICAL STOP AND CONTROL VALVES FOR TURBINES FOR HIGH PRESSURES AND TEMPERATURES

Fig. 28 shows three typical circular flange joints: First, that used on stop-valve bonnets; second, the type commonly used on control-valve bonnets; and third, as used in pipe runs where welding is undesirable because of dismantling requirements.

[18] "The Distribution of Load on the Threads of Screws," by J. N. Goodier, *Journal of Applied Mechanics*, vol. 7, March, 1940, p. A-10.

These joints all use a narrow, thin, soft iron enclosed gasket. The gasket is usually made about equal to the net cross-sectional area of the bolting. It is desirable oftentimes both to reduce the bolt circle diameter and to put in as many bolts as possible; the new cylindrical nut with the hex on top permits this.

The bolting shown on the stop-valve bonnet embodies a unique feature in that washers of about the same depth as the bolt diameter are used under the nuts. These have a cross section equal to that of the studs at the roots of the threads. The elastic system of washer and stud then has about three times the flexibility it would have if the nut rested directly upon the valve bonnet. This has proved useful in keeping the joint tight under conditions of a violent drop in temperature such as might follow priming of the boiler, and its use might advantageously be extended to other types of joints, but so far the need has not been apparent.

These joints have been successfully used for some time at pressures up to 2000 psi and 925 F, and practically no leakage difficulties have developed.

Valves

Fig. 29 shows typical control and stop valves used on large high-pressure high-temperature machines over a number of years. The streamlined or venturi-type valve was tested and first introduced about 1926, and has been refined in design and construction and applied to the entire line of turbines since then.

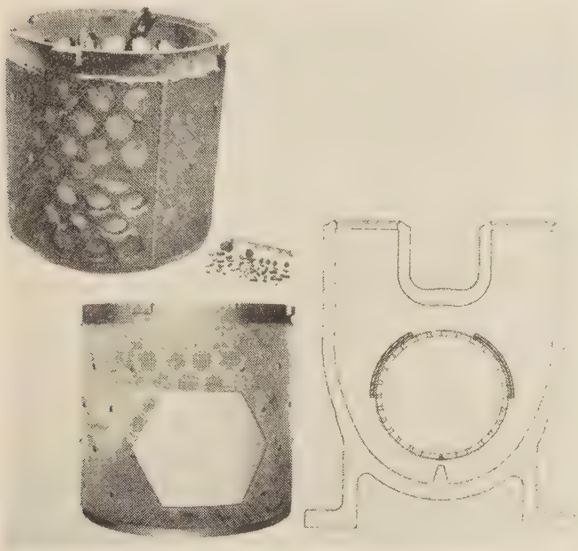

Fig. 30 Old and New Strainer Design

The seat is a rounded-entrance venturi tube, of great mechanical rigidity, and shaped so as to regain a substantial portion of the velocity-head drop through the opening. The valve proper is made a stream-flow shape, but that portion which engages the seat when closed is made a portion of a sphere, so that when closed it is as though a ball were dropped in a hole. Line contact is made with great pressure on the joint and inherent tightness results without grinding. If the valve should be injured it can be made tight again by remachining or stoning to a surface of revolution. The smooth flow at partial opening seems to be conducive to the maintenance to the valve surface, and special materials do not seem to be required.

Stuffing boxes in the older sense of the term have been eliminated and close-clearance metal bushings are used with one or two leak-offs. Nitrided stems and bushings are used up to 950 F, and there is some experience to indicate that this material may be satisfactory above this point, although research work has been under way for some time to determine the properties of other materials at these higher temperatures for this application. Hardness seems to be one of the essentials.

Experience over several years with more than a thousand valves of this type in service indicates that the difficulties usually associated with such parts have been reduced now to a small fraction of those previously encountered.

Strainers

Strainers, although they are seldom-thought-of parts of a turbine, play a most important function in keeping harmful foreign material out of the turbine blading system. Fig. 30 shows a strainer which has been battered and broken by small portions of material, probably welding wire left in the piping or superheater. In the older type of strainer this material was free to circulate around the outside of the strainer and to be blown back against the wire by the incoming steam, probably millions of times, until finally the wire was simply peened "in two" on the edge of the backing-plate holes opposite the incoming steam pipes. In the new design the strainer is made solid at this point, and a dam is placed in the chamber outside the strainer so that recirculation is not possible, and the particles will be trapped. No strainers of the new type have been battered through.

Hydraulic Servo-Mechanisms to Actuate Control Valves

The forces required to operate the control valve on large turbines are very great, although generally less on high-pressure machines than if the pressures were lower and other things equal. On the other hand, the tendency toward single-seat, streamlined, and hence unbalanced valves, together with the greater speed of operation required to prevent over-speed on loss of load by the high-output, light-rotor, 3600-rpm turbines, has aggravated this problem, and increased the power which the hydraulic valve-moving mechanism must have.

One method of meeting this demand for more hydraulic power has been to raise the operating oil pressure from the usual 125 psi to 250 psi, since by so doing greater output can be obtained in proportion to the energy put into the pilot valve by the governor. Another has been to relay the power of the centrifugal governor through a separate oil cylinder as has been common practice on governors for hydraulic turbines.

A typical hydraulic mechanism on a large turbine is shown in Fig. 31. In modern turbines this mechanism is completely enclosed in either the oiltight turbine bearing pedestal or in the oil tank under the front end of the turbine. In this way the oil fire hazard incident to exposed high-pressure oil piping has been made negligible.

The two-diameter piston is helpful in increasing the speed with which the valves can be closed, and is possible since the forces required to close the valves are less than half those required to open them.

Fig. 32 shows in diagrammatic form the governor and control system of a typical turbine of this type. Two important and rather recent aids to good operation are shown. The first is the hand wheel for operating the control valves in starting shown at *A*. The throttle valve has become a stop valve and has no intermediate setting between open and closed. In order to start the machine the hand wheel *A* is set in the closed position, and the stop valve opened to admit steam to the valve chest. The turbine can then be started on the control valves by turning wheel *A* toward the open position. Incidentally, steam is saved during the start, and the exhaust hood of the turbine heated more gently. When up to speed the speed governor takes over, but cannot open the control valves to a position greater than

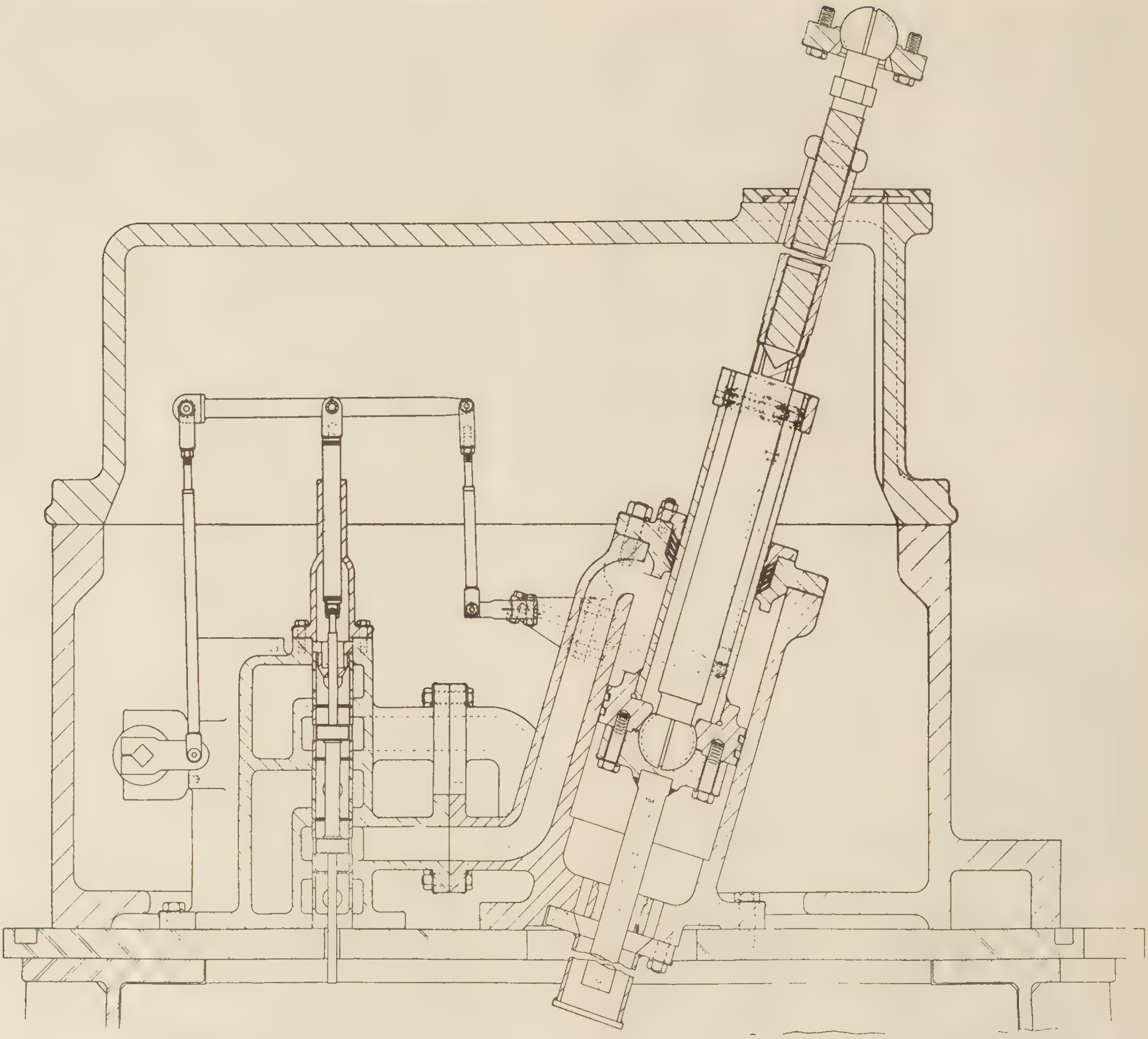

Fig. 31 Typical Hydraulic Mechanism Showing Two-Diameter Operating Piston to Secure Quick Valve Closing, Also Complete Enclosure of Parts Carrying High-Pressure Oil to Reduce Fire Hazard

that for which wheel *A* is set. Thus, the starting wheel becomes a valuable "load limit" which may be used to set the maximum load that a turbine may suddenly take on so as to protect boilers and other parts, or it may be used to set the load at any desired value for base-load operation of a particular turbine.

The second new device is called, for lack of a better name perhaps, the "magnetic pull-out." It is the invention of West Coast utility engineers, and has been applied on turbines recently delivered to one of these companies. In some systems connected with water power a major problem in steam-turbine operation is to operate the steam turbines safely as "spinning reserve;" i.e., to carry as little load on them as possible, but yet have them able instantly to pick up what load is required by an emergency. If the load is set down to the desired low value on the ordinary governor an increase in system frequency will cause the steam valves to be closed, the generator to "motor," and, hence, the turbine may overheat. If a fixed by-pass is put in to prevent this, it may jeopardize the ability of the operating and emergency governor to control the turbine overspeed on loss of heavy load. This new device is an adjustable magnetic link by which the governor may be set at any desired position, and then pulled down to the desired light load by the adjustable magnetic link. Any slight increase in frequency will have no effect, but a predetermined drop in frequency, which indicates need for this machine, will cause the governor to break the magnetic link, and the turbine will immediately take on whatever load the governor and frequency demand or the load limit may be set for.

Large turbines when operating as such spinning reserve have picked up load successfully from $^1/_{20}$ to full load in one minute's time. Proper precautions were taken to see that dry steam was supplied during the process.

Oil Piping

The general precautions taken against oil fires are: (1) To make all parts carrying oil under pressure of steel, with every possible precaution taken to insure against breakage or leaks; and (2) in addition to enclose all oil pipes or other parts carrying

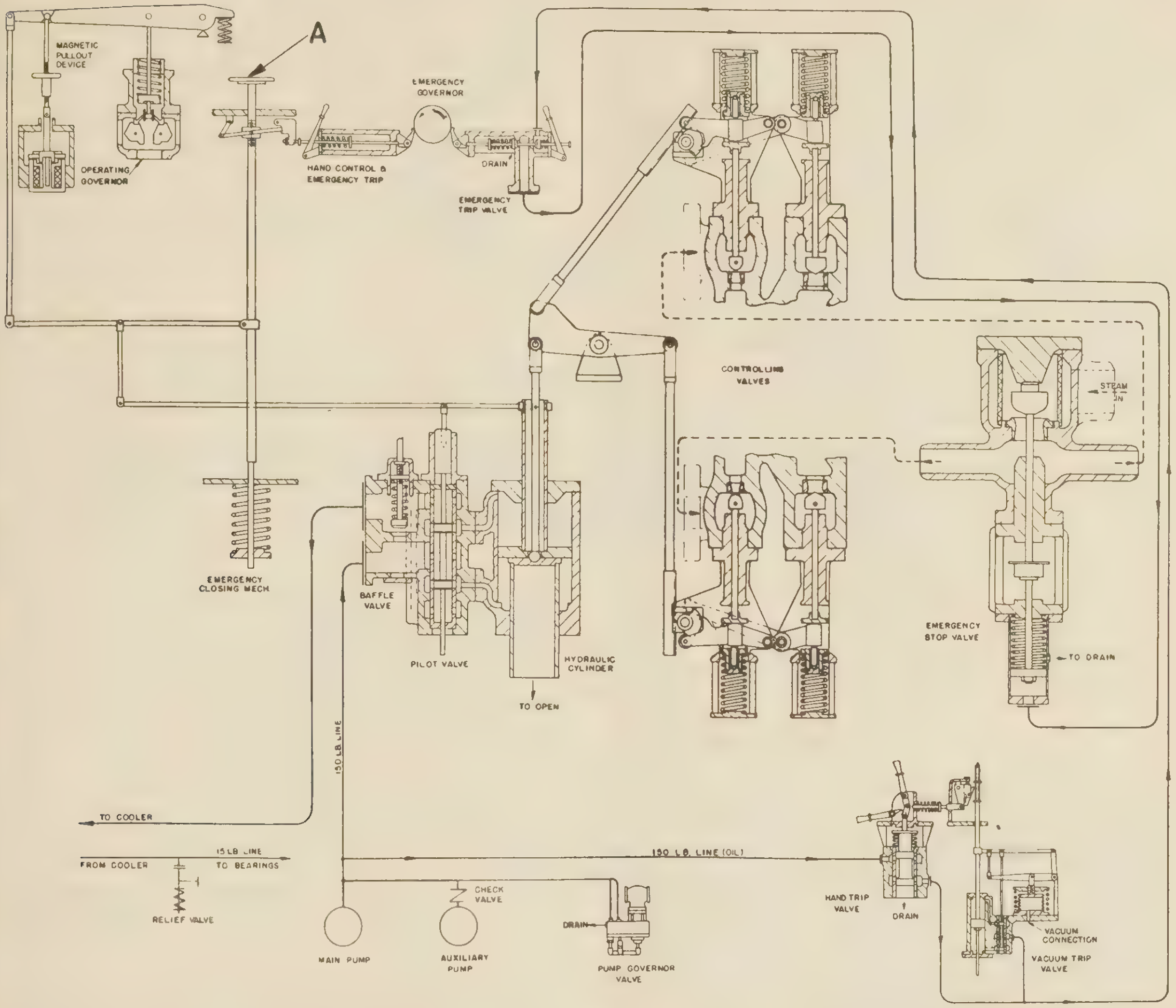

Fig. 32 Diagrammatic Sketch of Governor Mechanism Showing Handwheel *A* for Positioning of Main-Control Valve, "Magnetic Pull-Out" for Permitting Stable Light Load Operation, and "Vacuum Trip" to Shut Unit Down in Event of Loss of Vacuum

oil under pressure when adjacent to the high-temperature end of the turbine either within the oil tank, the turbine pedestals, or within the drain pipes. These seem to have been very effective in preventing oil fires.

Bearings

A very complete description of the "tapered-land" type of thrust bearing used was given in a recent paper.[19] The journal bearings frequently are operated at 220 psi pressure based on the projected area. Recent research work has made possible a considerable reduction in the air entrapped in the oil in its passage through the bearing which should have many favorable results.

Couplings

Experience has generally indicated that the flexible couplings, formerly used to connect turbine and generator rotors, are not only unnecessary, but also undesirable from the operation and maintenance viewpoints. It is now common practice to connect the rotors solidly together from one end to the other, with but one thrust bearing. Certain tandem-compound machines are exceptions to this rule. With correct bearing support the stability of operation is enhanced; the first cost and the maintenance costs are both reduced when compared to the use of flexible couplings.

Packings

Fig. 33 shows cross sections through both typical high-pressure and diaphragm packings on a modern turbine. The rings are cut in four to six segments and are held in grooves usually cut in the shell or diaphragm. Each packing segment is backed by a flat plate spring of a special high-temperature alloy which has been found to hold its tension. The shoulder in the groove prevents the segments' being pushed against the shaft. Up to about 850 F the rings are cut from a special centrifugally cast lead bronze, which has been found to have a low coefficient of friction when it comes in contact with the shaft, and to rub away readily. Above this temperature ribbons of pure nickel are set into a steel backing ring.

[19] "Thrust Bearings," by F. C. Linn and R. Sheppard, Trans. A.S.M.E., vol. 60, 1938, pp. 245–252.

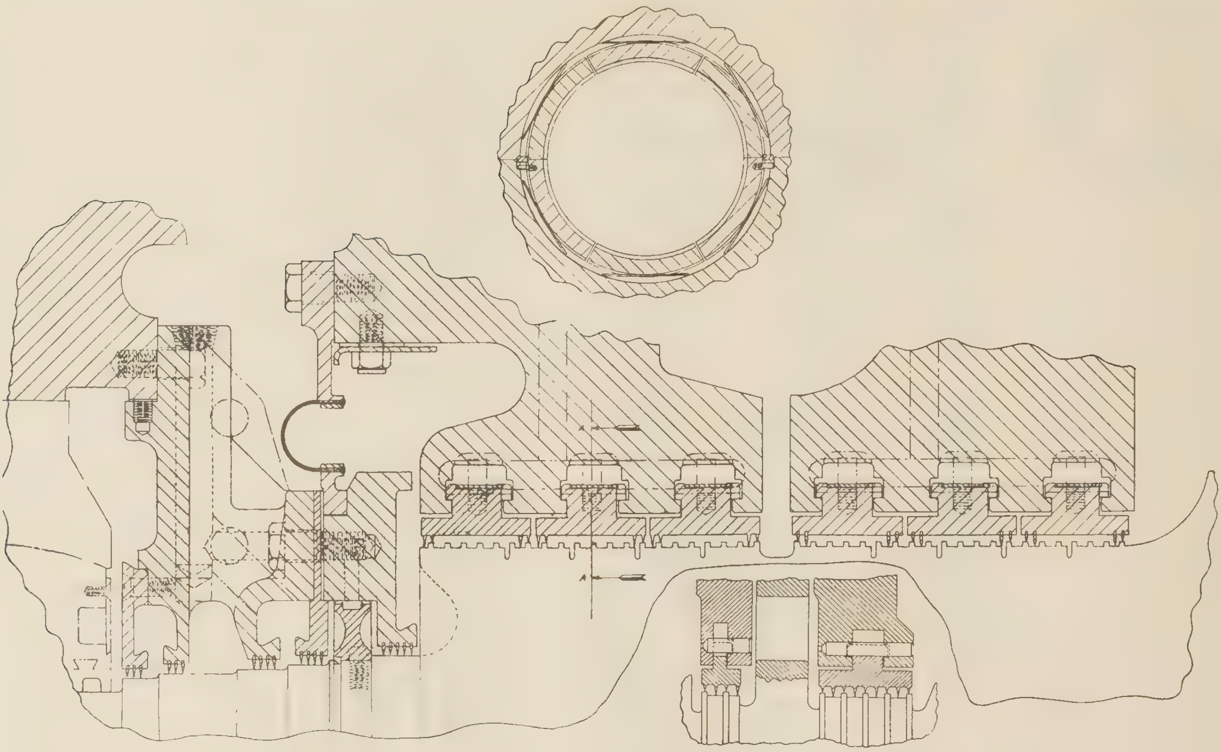

Fig. 33 Typical High-Pressure Shaft and Diaphragm Packings

The outer seal on both ends is a water-seal packing of the usual type.

This general type of packing has been in use on hundreds of turbines for several years, and has been by far the most successful type of packing which has been used. Its stationary elements are readily pushed out of the way by a crooked shaft on starting, and, if rubbed to a large clearance at any time, can be readily reset, on opening the turbine, to the original clearance, or renewed if necessary.

Turning Gears

All modern large turbines are now equipped with turning gears for the purpose of keeping the rotors rotating slowly during warming up, during cooling off, and generally during shutdown periods extending up to several days when it is expected that the turbine may be started again.

These turning gears have been of such great value that about 80 have been sold for installation on older turbines. If the packings are refitted when this is done, a very appreciable gain in economy is made.

Fig. 34 shows, in general terms, the reduction in starting time that can be made with the use of a turning gear. Actual times cannot be given because they vary so much from turbine to turbine and for different conditions, but it is believed that generally the relationships are correct.

It has not been found necessary to use high-pressure oil to lift the journals off their bearings when using a turning gear. A low-pressure motor-driven oil pump is used which floods the bearings with about half their usual flow of oil through the regular oil piping. The turning gear is then made powerful enough to start the rotor and rotate it at from $1^1/_2$ to 3 rpm. Sufficient lubrication seems to reach the bearing surfaces to prevent damage to journals. A brightening of the babbitt in the bottom of the bearings generally results.

Materials

This subject is both too broad and too specialized, as well as too controversial, for more than a brief discussion here. A great amount of research work is being carried out all over the world pertaining to the properties of materials for use at high temperatures.[20] Much original research work has also been done by the

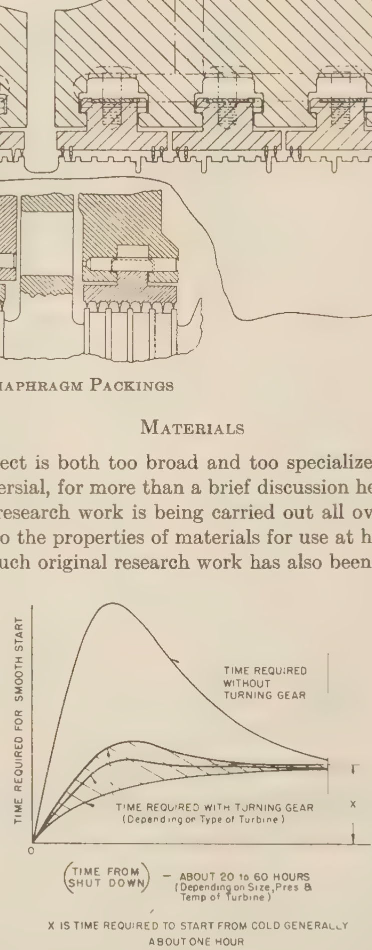

Fig. 34 Reduction in Starting Time Made Possible by Operation of Turning Gears

laboratories and the turbine engineering departments of the General Electric Company pertaining to turbine and power-plant materials. As rapidly as possible this is being shared with the many other workers in this important field, not only for their

[20] "Compilation of Available High-Temperature Creep Characteristics of Metals and Alloys." Compiled by Creep Data Section of Joint Research Committee on Effect of Temperature on the Properties of Metals (Joint Committee of A.S.M.E. and A.S.T.M.), March, 1938.

help, but primarily for their critical examination and discussion.[21]

Out of all this work on materials for higher temperatures have come a few generally accepted principles:

1 The addition of $^1/_2$ per cent to 1 per cent molybdenum is desirable for low-cost materials that will withstand temperatures up to 1000 F.

2 On these steels certain grain sizes, obtained by heat-treatment, seem to be desirable at the higher temperatures.

3 Elimination of "duplex" and dendritic microstructure.

4 Avoidance of use of aluminum in the refining process seems very desirable.

Research work started several years ago, first at the University of Michigan,[22] and somewhat later at the General Electric Company,[23] has introduced a new conception, however, as to the conditions necessary for the safe design of parts subjected to high temperatures.

Generally, designs for high temperature were predicated upon the "creep rate," and it was thought that if this was kept within certain values safety was assured. Furthermore, it was generally thought that the plastic flow introduced by creep would safely take care of stress concentrations which might exist. These later tests have indicated that rupture of parts subjected to high temperatures and stress follows rather definite time-temperature-stress relationships, and that strains which formerly appeared satisfactory may no longer be so if the parts are to have a useful life of, say, 100,000 hours. Further, these rupture tests indicate that materials when operating at high temperatures must be handled much as "brittle" materials like cast iron are treated at lower temperatures; i.e., concentrations of stress should be avoided as far as possible.

One result, for instance, of this latter viewpoint is that, whereas creep considerations alone indicated that cold straining of a piping system might not be necessary since "creep would soon relieve it anyway," these slow tests to rupture indicate that if much strain is necessitated by slow adjustments to some localized overstressed conditions, cracks might be started. This could be guarded against by cold-straining the piping system nearly to its hot condition.

On the other hand, the result of all of this research work has been to lend assurance that when the stresses are kept within the present allowable standards determined for the temperatures and materials under consideration and for structures such as turbines in which the clearance considerations have always required very low creep allowances, safe operation will be assured. There is reason to believe that in general the stresses now in use at 900 F may be more conservative in relation to the materials used than were those used ten years ago at 750 F.

There are, however, in addition to obtaining further rupture data, three rather important aspects of the action of steam at high pressures and temperatures on materials which will have to be cleared up before we can go much further. These have to do with: (1) The fatigue properties of metals; (2) the resistance to oxidation; and (3) the possible weakening effects of structural changes brought about by the surrounding conditions. Studies and investigations which are under way and in preparation with respect to these factors together with the additional progress promised by the metallurgists may still further extend our limits of safe operation.

Conclusion

The steam-turbine power plant is the most efficient means which we now have commercially available for transforming the energy of our great solid-fuel resources into the power which plays such an important part in the lives of us all. The value and growth of the fuel-burning power plants will probably increase in the years to come with the complete utilization of our definitely limited water-power resources. The steam turbine has maintained its position of leadership by the ease with which it can be made to produce large quantities of power economically. The progress in turbine design and construction, some of which is outlined in this paper, has been a necessary part of the maintenance of this position, and the further progress which appears possible indicates that the steam turbine will probably maintain its leadership for a great many years to come.

Discussion

M. W. Benjamin.[24] Two matters of special interest are shown by the several performance curves included in the paper. The first is that turbine designers are now able to predict performance with much greater certainty than in years past. As an example, one may compare the test and guarantee curves of Fig. 19*B*, with those of Figs. 20*B*, 20*C*, and 21. These curves show definitely the trend toward reducing the margin between guarantee and actual performance from a value of some 3 or 4 per cent a decade ago to around 0.5 to 1 per cent as of today. It is of value in designing a plant to know that turbine performance as guaranteed may be counted upon so accurately, since it permits closer calculations on optional investments for improved plant efficiency.

A second point of interest in the paper is presented in Figs. 20*C* and 21, which show that the most satisfactory choice of throttle steam conditions depends not only upon fuel costs and loading conditions but also upon the plant designers particular preference as to how the available money is to be spent. In Fig. 20*C*, a 65,000-kw tandem-compound hydrogen-cooled machine, operating on 650-psi steam, has a heat rate about 200 Btu per kwhr lower than that of a 75,000-kw single-casing air-cooled machine, operating at 815 psi gage. Exhaust pressures, throttle temperatures, feedwater temperatures, and number of feed-heating stages are essentially the same for both units. As between the tandem-compound unit of Fig. 20*C*, and the 80,000-kw, 1250-psi unit of Fig. 21, the increase in pressure from 650 psi to 1250 psi produces only 150 Btu per kwhr improvement in turbine heat rate. As between the 60,000-kw 600-psi gage 825 F unit, the 75,000-kw 815-psi gage 900 F unit, and the 80,000-kw 1250-psi gage 900 F unit of Fig. 21, it is evident that two thirds of the improvement possible between 600 psi gage 825 F and 1250 psi gage 900 F can be obtained by increasing the pressure only one third of the way; that is, 215 psi, and the temperature 75 F. Thus the advantages

[21] "Flow of Steels at Elevated Temperature," by F. P. Coffin and T. H. Swisher, Trans. A.S.M.E., 1932, APM-54-6, p. 59.

"Stability of Steels Under Stress at Temperatures Up to 1000 F (as reviewed by a turbine designer)," by Ernest L. Robinson, *Metal Progress*, Sept., 1935, vol. 28, pp. 34–39, 78.

"The Creep of Steels as Influenced by Microstructure," by L. L. Wyman, *Mechanical Engineering*, vol. 57, 1935, pp. 625–627.

"Actual Grain Size Related to Creep Strength of Steels at Elevated Temperatures," by S. H. Weaver, Proc. A.S.T.M., vol. 38, Part II, 1938, pp. 176–196.

"Fracture of Steels at Elevated Temperatures After Prolonged Loading," by R. H. Thielemann and E. R. Parker, Trans. A.I.M.E., Class C, Iron and Steel Division, vol. 135, 1939, p. 559.

[22] "The Fracture of Carbon Steels at Elevated Temperatures," by A. E. White, C. L. Clark and R. L. Wilson, Trans. American Society for Metals, vol. 25, September, 1937.

"The Rupture Strength of Steels at Elevated Temperatures," by A. E. White, C. L. Clark, and R. L. Wilson, Trans. American Society for Metals, vol. 26, pp. 52–80, March, 1938.

[23] "Fracture of Steels at Elevated Temperatures After Prolonged Loading," by R. H. Thielemann and E. R. Parker, Trans. A.I.M.E., Class C, Iron and Steel Division, vol. 135, 1939, p. 559. Technical Paper No. 1034.

[24] Engineer, Engineering Division, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

of improved turbine design, low leaving loss and hydrogen cooling are in direct competition with improved performance obtained through increases in steam pressure.

Users of large turbines are grateful to the author for this paper. As it is restudied and re-examined the facts presented will take on added significance to the general benefit of all steam-power engineers.

C. B. Campbell.[25] Attention is directed to the author's statements regarding the desirability of adopting 3600-rpm turbine designs within their practical capacity range. The writer takes this opportunity to support his conclusion. With high steam pressures and temperatures, in particular, it seems axiomatic that the small high-speed unit should be superior to the 1800-rpm turbines both as to reliability and sustained high efficiency. Actual operating experience with 3600-rpm turbines exceeding 15,000-kw rating is limited to a relatively short period of time, but such conclusions as can be drawn to date are distinctly favorable.

The author refers to moisture erosion of high-tip-speed exhaust-end blading of condensing turbines. Centrifugal moisture removal and stellite shielding, combined with shrouded and airfoil-section-tipped blades has relegated this problem to one of secondary importance in modern turbines. But slight erosion has been found after 2 years of service with blades having tip speeds of 1256 fps in steam having a nominal moisture content of 12 to 13 per cent.

M. K. Drewry.[26] That modern turbines merit cleaner steam, and never any water, is certainly an appropriate admonition. An ill-advised standard of boiler-water treatment has and apparently will continue to cost the power industry enormous sums in unnecessary coal because of the carry-over it promotes. To achieve "chemically pure" turbine blading is worth much effort.

That turbine manufacturers must provide for large quantities of water ("priming") in high-temperature high-pressure turbines is not creditable to boiler design or boiler operation. For 900 F turbines to receive "shots" of water should be considered as serious, for instance, as water in the generator. Boiler-water-level control continues to be a most important item in plant operation.

Arthur McCutchan.[27] The author's statement to the effect that the subject of materials for high-temperature service is controversial cannot be questioned, but the "few generally accepted principles" listed are certainly far from acceptance. These principles are discussed as follows:

1 The value of adding molybdenum to secure greater creep strength is well substantiated, although the addition of chromium also appears desirable from both the corrosion and strength standpoints, if temperatures as high as 1000 F are involved.[28]

2 It has been reasonably well established that a fairly large grain size obtained by control of melting practice is desirable. Attempts to secure large grain size by heat-treating naturally fine-grained steels have proved disappointing as far as creep resistance is concerned.[29]

[25] Manager, Land Turbine Engineering, Westinghouse Electric & Manufacturing Company, Philadelphia, Pa.

[26] Assistant Chief Engineer of Power Plants, Wisconsin Electric Power Company, Milwaukee, Wis. Mem. A.S.M.E.

[27] Engineer, Engineering Division, Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

[28] "High-Temperature-Steam Experience at Detroit," by R. M. Van Duzer, Jr., and Arthur McCutchan, Trans. A.S.M.E., vol. 61, 1939, pp. 392–396.

[29] "Investigation of Influence of Grain Size and Creep Strength of Carbon-Molybdenum Steels," University of Michigan project for The Detroit Edison Company.

3 The so-called dendritic structure observed in wrought molybdenum is otherwise known as "acicular" or Widmanstätten structure and is associated with the best creep resistance so far obtained in creep tests on molybdenum steels.[30] Instead of eliminating this structure, the present tendency is to try to retain it by substituting normalizing for the present annealing treatment after fabrication of grain-size-controlled molybdenum pipe. The effect of "duplex" grains on creep strength remains a disputed point among metallurgists.

4 In producing the present larger grain-size material, mill practice is to add 1 lb of aluminum per ton rather than 2 lb, as in former practice. This reduction can hardly be said to represent "avoidance of the use of aluminum."

The rupture tests referred to by the author demonstrate only what has been known for many years, namely, that the majority of materials, if slowly stretched at high temperatures, fail by intercrystalline rupture with little plastic deformation. It is misleading to imply that material becomes brittle or like cast iron under such loading, since the material, if rapidly tested at room temperature after a period of service at high temperature, will show practically its original ductility.

The relief of excessive bending stress through creep is a relaxation phenomenon similar to that encountered in bolting or relaxation-creep tests. The writer would like to know if the author has encountered failure under any type of relaxation loading. Because of the initial and fairly large adjustment which occurs in the first few hundred hours at temperature, there is little reason in the writer's opinion to fear any occurrence that could be remotely related to intercrystalline failure as far as bending stress is concerned.

T. C. Rathbone.[31] The chart Fig. 18 of the paper, representing for each year the average length of service of original rows of turbine buckets on which trouble was reported, is interesting. It is assumed that the figure for each year represents the mean time for the aggregate of all buckets which have given trouble during that year. Thus, for 1938, the figure of approximately 10 years is taken to represent the average of a number of cases, ranging from buckets that had been installed relatively recently to buckets that had been in service 20 or 30 years.

The definition of "trouble" probably includes erosion and corrosion as well as fatigue failure in either the bucket proper or its root. The interesting point is that fatigue failures can occur after so many years of operation. The popular conception of failure by fatigue involves stress reversals or fluctuations with intensities exceeding the appropriate endurance limit, and this limit is commonly defined as the value which the S-N curve approaches asymptotically, which is roughly after 10,000,000 or 15,000,000 cycles.

Assuming a nominal bucket-vibration frequency of 500 per sec and a turbine use factor of 8000 hr per year, the total number of bucket vibrations accumulated would be about 150,000,000,000. Peterson[32] has reported fatigue failures after 100,000,000 cycles, but some other explanation seems necessary to account for failures occasionally experienced where the stress cycles reach astronomic numbers.

Either some change has been brought about just prior to failure to alter the bucket frequency into a closer approach to the resonant condition, such as by changes in the mass of the bucket

[30] "Quick Determination of Limiting Creep Stress," letter by Walter Rosenhain, *Metal Progress*, Feb., 1932, pp. 65–66.

[31] Chief Engineer, Turbine and Machinery Division, The Fidelity and Casualty Company of New York, New York, N. Y. Mem. A.S.M.E.

[32] Research Department, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

by erosion, or in the root-fastening condition; or else the resonant amplitudes are built up only during some transient operating condition, such as at an infrequent partial load, or at some lower speed which is passed through only when the unit is taken out or placed in service. The critical condition may occur at an overspeed which is transited only on the occasion of the infrequent overspeed tests to check the emergency governor. Thus, years may be required to accumulate a sufficient number of cycles at stresses slightly above the endurance limit to bring about failure.

The curve, Fig. 18, represents only the bucket rows which have given trouble. It would be interesting to supplement this chart by means of a suitable ordinate to show the relation between rows in trouble and total rows in service. This would give a better perspective of the problem, and would emphasize the remarkably small percentage of buckets which have given trouble.

It is gratifying to note the number of older turbines that have been equipped with turning gear. These installations represent worth-while investments. In addition to the gains mentioned by the author, namely, the reduction of starting time and gain in economy, the turning gear also eliminates the hazard of rubbing and excessive vibration when the unit is brought to speed.

The first turning gears, in the development of which the writer participated, were designed to rotate the spindle about 25 rpm. Experiments[33] were made with an elaborate setup for recording low speeds accurately, to find the point on drifting down to standstill at which the bearing oil films first began to break down. This was determined by the break in the deceleration curve. Values from 10 to 18 rpm were found, depending upon the oil and temperature. A speed of 25 rpm was selected as being safely above this point. Some fear was then entertained that wiping might occur at lower speeds.

The disadvantage of the gears for this speed was that they had to be engaged at exactly the proper moment on drifting down, as the spindles could not be started from a standstill by the gear, unless an auxiliary high-pressure bearing oil system was installed. Otherwise, it was necessary to turn over the rotor with steam.

It was later found that turning gears could be operated at much lower speeds, apparently with no difficulty from wiping, and gears turning the spindle at only 1 or 2 rpm are able to start rolling from a standstill without high-pressure oil.

When turning gears on central-station turbines were first introduced, the urge to eliminate the starting-up rubbing and damage to packing and blade tips caused by distortions was about as vital as the desire to reduce the starting-up time after a short shutdown. When the unit is standing hot after a shutdown the stratification of hot gases, collecting at the top of the casing distorts both the rotor and the cylinder into an upward bow, reaching a maximum in say 8 hr.

If the distorted rotor is turned over $^1/_2$ revolution in this condition, the radial clearance, between its parts and the upward-distorted cylinder parts is at a minimum. Cylinder distortion although of secondary importance may yet be of concern.

At 25 rpm, the fanning action of the blades tends to whip the gases around the cylinder and prevent cylinder distortion. At 1 to 2 rpm, no such benefit would seem to be possible. The writer would like to inquire whether there has been any experience indicating difficulty from cylinder distortions with the slow-speed gear?

A. M. Selvey.[34] The problem of moisture in the lower turbine stages becomes more and more important with every increase in throttle steam pressure. It is well known that moisture has two detrimental effects on turbine performance (1) the erosion of buckets, and (2) the reduction in stage efficiency. Because structural materials for turbine buckets are not yet available which will withstand the action of higher exhaust-moisture contents, the turbine designer must limit them to about 12 to 15 per cent. This situation is an appreciable handicap since, with the high stage efficiency and high steam pressure and temperature obtaining today, up to 20 per cent moisture could be formed within the turbine with a large attendant increase in over-all efficiency, despite the reduction in stage efficiency due to the extra moisture. While there is a small loss of efficiency, due to the braking effect of the extra 5 to 7 per cent of moisture present, it represents only 4 to 5 per cent of the additional energy liberated by the moisture formation. Every 1 per cent of moisture formed in the turbine releases approximately 10 Btu per lb of steam flow.

With the publication of W. M. Meijer's recent paper, on "The Extraction of Condensate From Expanding Steam,"[35] the attractive possibilities of moisture withdrawal from the lower turbine stages was again brought to notice. Turbine designers have been striving for many years to develop and perfect convenient mechanical means of moisture separation, preferably without cumbersome equipment external to the turbine. With the accomplishment of this desirable operation, it will become more practicable for power-plant designers to increase throttle steam pressure and temperature. Until such time, there is slight advantage in adopting high steam pressure and temperature, if turbine efficiency must be sacrificed to keep exhaust moisture within bounds prescribed by bucket erosion.

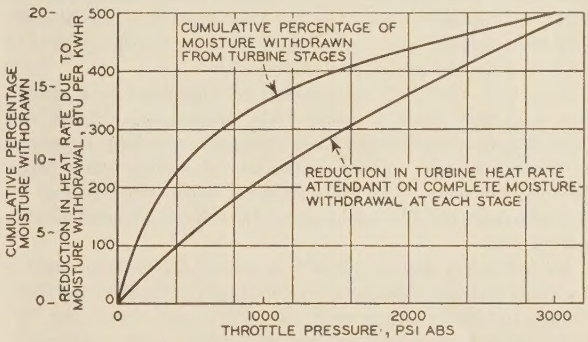

Fig. 35 Maximum Betterment in Turbine Performance Attainable Through Moisture Withdrawal

(Infinite number of feedwater heaters and stages of boiler feed pumps; regeneration to throttle saturation temperature; zero heater-terminal difference; condenser pressure, 1 in. Hg; no generator or mechanical losses; dry-stage efficiency 84.5 per cent; throttle temperature, 900 F.)

Fig. 35 of this discussion shows the maximum reduction in turbine heat rate obtainable by complete moisture withdrawal for the throttle conditions and cycle noted. The cumulative total amount of moisture withdrawn also is indicated. To obtain an equivalent of the 300-Btu per kwhr reduction in turbine heat rate, attendant upon complete moisture withdrawal at 1500 psi abs, the throttle steam pressure would have to be increased to 2600 psi abs. It is understood that considerable benefit in reduced heat rate obtains even when moisture withdrawal is only partial and not complete. It should be pointed out that, when moisture withdrawal and increased steam pressure go hand-in-hand, sizable cumulative saving may be realized.

It is to be hoped that Dr. Meijer's paper will quicken the in-

[33] "Turbine-Shaft Distortion Corrected by Spindle Rotation," by T. C. Rathbone, *Electric Journal*, vol. 28, Feb., 1931, pp. 91–95.

[34] Engineer, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

[35] "The Extraction of Condensate From Expanding Steam," by W. M. Meijer, *Journal of the Institution of Naval Architects*, London, vol. 81, 1939, pp. 36–48. See also *Engineering*, p. 416, April 7, 1939, and *Combustion*, August, 1939.

terest of turbine manufacturers in the possibility of moisture withdrawal to the end that they will actively attack the problem with a view to more fully realizing the advantages to be gained therefrom.

Philip Sporn.[36] This is not only an excellent paper, broad in scope, but also an excellent answer to those who either bemoan lack of opportunity for making further progress in the power-production field or who would prefer to go back to the days in power-plant design when pressures and temperatures were fixed at 200 lb and 550 F. It is also an answer to those individuals who, without the necessary knowledge, take a set of unrelated figures and attempt to prove that the summation of a series of coordinated progressive steps constitutes a retrogression.

In part 1 of the paper, Figs. 2 and 3, which show so well how the large 3600-rpm turbine and hydrogen-cooled generator have aided the cause for higher pressure and temperature (and plant performance), are noteworthy. Perhaps few appreciate that "it is doubtful if it would be practical to design a 1200-rpm turbine of reasonable efficiency to operate at 1200 lb pressure 950 F." Perhaps it is not common knowledge that large 3600-rpm generators of over 25,000 kw capacity were not available before 1935.

In part 2, a striking use of comparative illustrations indicates the development of the double shell. The top cross section of Fig. 8 shows the front end of the Logan 40,000-kw turbine,[10] the first turbine to have a double shell. Below is shown the 60,000-kw Windsor turbine, the first to have the "valve-in-head" design. When each of these turbines was opened, the close clearances found to have been maintained by the diaphragm and shaft packings were remarkable. The bottom cross section shows the 25,000-kw turbine for the Missouri Avenue plant of the Atlantic City Electric Company. All of these units are on the systems of the American Gas and Electric Company.

Since it is "details" which are most important in a design, part 3 of the paper is particularly interesting. Piping and valve designers could well take advantage of the ideas expressed on bolting threads, as well as the use of washers under stop-valve-bonnet bolts. The replacement of valve-stem packings by close-clearance metallic bushings might well be extended to more general use.

An additional special "detail" which might be mentioned is the initial pressure regulators developed for the Windsor and Twin Branch turbines which protect the boilers (and, hence, the turbines) against too great a pressure drop which might cause carry-over and too rapid cooling of drums and other thick parts. We are also using such a regulator for the Atlantic City and Philo turbines.

A large number of the turbines described will soon come on the line or are already in operation. We hope that a paper will be presented at a future meeting summarizing actual operating experience with these, and new developments in design, resulting from that experience.

The author points out the stabilization of design speeds at 3600 rpm for some time to come. While this is so, we hope and believe that no stabilization will occur in other fields. Certainly, all should maintain the same spirit of inquiry and resourcefulness evidenced in this paper by the author and his associates. Such an attitude will assure not only more reliable but also more economical turbines.

Author's Closure

In reply to Mr. Benjamin's comments, it would be misleading if the results of the tests shown in Figs. 19 and 20 were compared with the guarantees shown in Fig. 21, particularly for such small differences as the 1 and 2 per cent differences mentioned by Mr. Benjamin. A companion paper presented at the same meeting jointly with Mr. Knowlton[37] is designed to permit a more definite appraisal of the relative fuel consumptions as between turbines of comparable design for different steam conditions, with and without hydrogen cooling and with different leaving losses.

It is, of course, true that the improvements due to tandem compounding or other improved turbine-design features, low leaving loss, and hydrogen cooling might be considered as in competition with comparable gains which can be obtained by higher steam conditions. However, it would seem that it might be better to consider that on the whole these might be used to supplement each other.

Supporting Mr. Campbell's statement that the erosion problem on the last-stage buckets has been relegated to "one of secondary importance in modern turbines," it might be of interest to state that four last-stage buckets of airfoil section at the tip have been in operation at 1201 fps with a theoretical moisture content of 13 to 14 per cent for an average of $8^1/_2$ years and without sufficient erosion to warrant replacing, or any serious reduction in efficiency. These blades did not have stellite erosion shields.

The writer can only indorse the stand taken by Mr. Drewry with respect to the desirability of keeping water out of a turbine. However, the fact remains that this has happened in a number of cases in the past, even with relatively modern steam conditions and boilers, and it has been our intention to do everything possible in connection with the design of the turbine to minimize the detrimental effects if it does happen. It is quite probable that a turbine designed with these drastic conditions in mind, if it can be designed so as to withstand such severe service, will be a better turbine to withstand the much less severe but ordinary variations in temperature incident to rapid load changes and rapid starts.

Making allowance for a somewhat different use of words, the author finds himself for the most part in agreement with the comments offered by Mr. McCutchan. Indeed, his discussion constitutes an amplification of the author's very brief review of the principles controlling high-temperature strength.

1 Chromium was not mentioned by the author because its beneficial characteristics are effective throughout a wide range of temperatures.

2 The author welcomes Mr. McCutchan's more complete statement about grain size. As a matter of fact, any adequate discussion of this subject might well cover an entire article by itself.

3 The objectionable dendritic microstructure referred to by the author might perhaps better have been called dendritic segregation or banding. It is visible to the naked eye and is altogether different from the acicular microstructure which gives high values of creep strength in ordinary creep tests. There is no disagreement about these matters once the meaning of the terms is cleared up. By duplex microstructure the author referred to the presence of large grains and small grains at the same time, and he is not aware that any metallurgist regards such an arrangement as desirable.

4 The author agrees with Mr. McCutchan that a reduction of aluminum addition from 2 to 1 lb has not gone very far in this direction. The present evidence is that such mills as still make these large additions of aluminum can improve the high-tem-

[36] Vice-President and Chief Engineer, American Gas and Electric Service Corporation, New York, N. Y. Mem. A.S.M.E.

[37] "Relative 'Engine Efficiencies' Realizable From Large Modern Steam-Turbine-Generator Units," by G. B. Warren and P. H. Knowlton, presented at the Semi-Annual Meeting, June 17–20, 1940, of The American Society of Mechanical Engineers.

perature quality of their product by further alteration of their deoxidation practice.

Mr. McCutchan's remarks with reference to the rupture test are welcome. While it is true that the intercrystalline character of such rupture has been known for a long time, the discovery of a definite stress-time relationship whereby it is possible to assign working stresses with a definite margin of safety dates from the first paper on this subject presented in 1936 by C. L. Clark[22] of the University of Michigan and published the following year.

The author did not say that the material becomes brittle, but that it "must be handled much as brittle materials." In reply to Mr. McCutchan's question, the author would say that no actual failures have been encountered in service under any type of relaxation loading, although it is necessary to note that in a few cases bolts have had to be replaced after repeated dismantling and reassembly. The author hopes that Mr. McCutchan's optimism with reference to bending behavior is justified.

Mr. Rathbone's analysis of the situation leading to bucket failures after 10 years or more of operation is probably correct. The definition of trouble does include erosion and corrosion as well as fatigue failure in either the bucket proper, its root, the shroud band, or the tie wire. The bulk of the buckets in trouble have been those designed before the method of proportioning described in the paper was developed and applied. The number of buckets which have given trouble is only a small fraction of the total in service. Upon careful consideration at the time the paper was written it was thought that the presentation of the information contained in Fig. 17 was a better presentation of this matter than any records of our own would be.

With respect to Mr. Rathbone's statements regarding the turning gears, the question of the proper speed for turning gears has been a matter of discussion for years. The low speed adopted, as Mr. Rathbone states, has been primarily because of the greater simplicity in design and operation attainable with the lower speeds. So far as the author is aware, there have never been any difficulties attributable to insufficient speed of the turning gears on the types of turbines described in the paper, that is, on turbines with relatively short shafts and shells, with large radial clearances at the bucket tips and with packings having spring backing. Packing measurements on some machines during inspection have indicated slightly more clearance at the bottom of the diaphragms in the middle of the shaft length than on the sides or top, such as might be due to an upward bow in the casing, and this might indicate the desirability of a higher turning gear speed. It is probable that this will be one of the developments of the next few years.

As pointed out by Mr. Selvey, turbine designers have devoted much attention for many years to constructions which will drain moisture from the various turbine stages or cross-over connections during operation. The gains are, of course, very substantial but must be balanced against the increased capital costs which would be required, particularly if attempts are made to extract the moisture from cross-over connections between turbine casings or, if the turbines were divided into various casings to permit extraction of moisture from the pipe connections in between. One means of eliminating the moisture which was quite commonly used a few years ago, before the advent of higher initial temperatures, was resuperheating. There is much evidence to indicate that resuperheating may have definite economic advantages even with modern high initial temperatures. As pointed out by Mr. Campbell, erosion is not a major problem in modern turbines, and it has never been necessary with steam conditions now in use to sacrifice turbine efficiency "to keep exhaust moisture within bounds prescribed by bucket erosion."

Mr. Sporn's comments regarding the value of the paper are appreciated. No reference to the initial-pressure regulator was made in the paper because at the time the paper was in preparation this device was in the process of development with Mr. Sporn's organization. However, this regulator was recently described in an article[38] in the technical press.

[38] "Recent Development in Turbine Governing to Meet Special Conditions," by R. J. Caughey, *Combustion*, June, 1940, vol. 11, pp. 27–29.